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Mechanical and Refrigerating Engineers' Handy-Book

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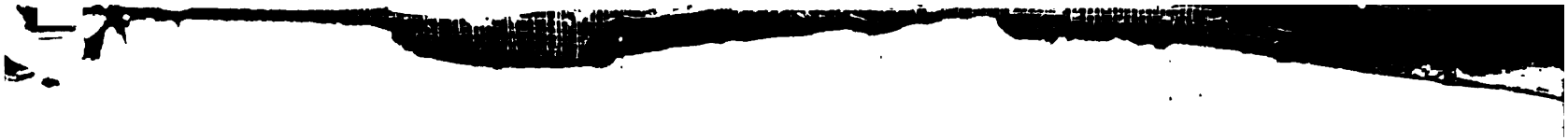

Mechanical and Refrigerating Engineers' Handy Book

BY

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Preface.

In writing this book, the author has had in mind the preparation of a work which would serve as a handy reference for the practical engineer, particularly the refrigerating engineer. The book, however, is also designed to supply the needs of the practical steam engineer.

The information contained in the book has been collected from numerous sources, and considerable of it is based on the author's own practical experience. The field covered is a very wide one, and not a little work has been devoted to making it thorough. It is realized that in a first edition of a book of this type, where there is little precedent to follow, there are liable to be omissions and errors, and notices of such from the users of the book will be appreciated.

Considerable care has been given to the tables in the book. Many new tables have been compiled, and old tables carefully gone over and corrected. The latest Steam Table by Marks & Davis, and the newest Ammonia Table by H. J. MacIntire have been inserted and adopted as the standard for the book.

The policy of giving numerous calculations, all illustrated by examples, has been adhered to throughout the work. This is done to give any practical engineer the opportunity to calculate them correctly himself. The various formulas given are in general written according to the standard method, that is where a multiplication sign appears, the operation indicated by this sign must first be performed.

The author has had the kind assistance of various scientists and engineers, by whose help many valuable data have been obtained, and to whom thanks and appreciation are hereby

cordially extended. He is under special obligation to E. T. Henius, Expert, Wahl-Henius Institute, for his able editorial work; to G. Thevenot, formerly Expert with the Wahl-Henius Institute, for work in connection with the revision of the manuscript; to G. F. Gebhardt, Professor of Steam Engineering, Armour Institute, for numerous suggestions; to G. L. Goob, Expert, Wahl-Henius Institute, for the preparation of the chapters on "Boiler Water and its Treatment" and "Friction and Lubricants;" to James Banash, Engineer, Chicago, for the preparation of the chapter "Internal Combustion Engines;" to G. J. Patitz, Chief Engineer, Schoenhofen Brewing Co., Chicago, for work on the chapter "Electrical Engineering" and various suggestions on other chapters; to N. L. Feilberg, formerly with Wahl-Henius Institute, for many of the drawings, and work on manuscript; to O. Roewade, Chief Chemist, Wahl-Henius Institute, for the preparation of the chapter "Chemistry." The author also wishes to thank the various manufacturers who have furnished cuts and information for the book.

OTTO LUHR.

August, 1913.

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Physics

Introduction.

Matter is anything that occupies space, including such substances as earth, metals, water, air and other gases, etc. A limited portion of matter is a body.

While the science of physics does not consider the chemical composition of matter nor any of the phenomena of life, it deals with all other properties of the various kinds of matter, especially with the forces that arise, tending to alter the position of a body, to change its shape and size, or to change it into smaller bodies, and with the resistances acting against these forces. Therefore it may be said, that physics is the science of matter and energy.

A lump of sulphur can be reduced to a powder by grinding, melted by careful heating, made electric by rubbing with a woolen cloth, but after all these changes, still remains sulphur, and such changes, therefore, belong to the domain of *physics.* However, if sulphur is ignited, it burns with a pale blue flame, emitting a suffocating odor, disappears, and is no longer sulphur, a new substance having been formed. Such a change is not a physical, but a chemical change, and the study of such a change belongs to the science of *chemistry.*

States of Matter.

Matter occurs in three states, *solid, liquid* and *gaseous.* Many substances are met with in nature which exist in all three of these states. Liquid water, for instance, changes under certain conditions into a solid, ice, and under other conditions into a gas, and is then called steam. With sufficiently high temperatures, all substances will take the form of a gas.

Solids are bodies that have a strong tendency to keep the same shape and the same volume in all positions.

Liquids are bodies whose shapes depend on the vessels that contain them, but they retain a definite volume.

Gases are bodies that have a tendency to occupy as large a space as possible.

Molecules.—The fact that all bodies expand by heat, and that they can be changed from solids to liquids and gases, can be easily explained, if we suppose that all bodies are composed of very small, separate, and movable particles. To these small particles the name molecules has been given. A molecule is so extremely small that the most minute particle of dust floating in the air contains millions of molecules.

We suppose that the molecules are held together by a certain force called *cohesion.* This force is strong enough in solid bodies to keep the molecules together. In liquids, cohesion is weaker and allows the molecules a freer movement. In the gases, the molecules being far apart, cohesion entirely disappears.

Properties of Matter.

Extension.—Matter occupies space; it is measured by its length, breadth, and thickness, called the three *dimensions.* This property or quality of matter is expressed by the word extension.

Mass and Weight.—The amount of matter of a body is expressed either as volume, weight, or mass (for explanation of the two latter expressions, see page 16).

Impenetrability.—The fact that two bodies cannot be in the same place at the same time is expressed by the term impenetrability of matter. If for instance an inverted bottle is immersed in water, the water, on account of the presence of air in the bottle, cannot rise into the bottle to fill it.

Indestructibility of matter signifies that matter cannot be destroyed. The total amount of matter in the universe cannot be decreased.

Inertia is the tendency of matter to remain in the same condition, that is, if at rest, to remain so, and if in motion, to continue to move with unchanged speed in the same direction.

Elasticity is the property of matter to resist, within certain limits, pressure, pulling, bending or twisting, and to resume its original form when the force ceases to act upon it.

Porosity.—As all bodies are composed of separate small particles, or molecules, they only apparently fill the space they seem to occupy. The interstices between the molecules are called *pores,* and bodies are said to have porosity. If, for instance, a tumbler is entirely filled with water, some alcohol can be poured into the water without overflowing the tumbler, because the alcohol enters into the spaces between the molecules of the water.

Hardness.—When one body is able to scratch another, it is said to be harder than the other. For instance, a piece of steel will scratch a piece of copper, and hence is said to be harder.

Tenacity is the quality of matter to resist a force tending to pull its particles apart from each other.

Malleability.—Some bodies can be hammered or rolled into sheets, and such bodies are said to possess malleability. Iron and, to a still greater extent, gold, can be worked into very fine sheets.

Ductility.—Some bodies, for instance, glass, iron, platinum, can be drawn out into very fine wires, and such bodies are called ductile.

Crystallization.—When a solution in water of sugar, of salt, or of certain other substances, is allowed to dry up slowly, the solid bodies in solution often assume a regular shape, as a cube, a pyramid, etc., and such bodies are called *crystals.* Thus, salt crystallizes in cubes, alum in pyramids, sugar in prisms.

Absorption.—A few solid bodies have the power of taking up or absorbing gases. Thus, charcoal can condense in its pores as much as 90 times its own volume of ammonia gas.

Diffusion.—Some liquids, and all gases, when brought into mutual contact will penetrate each other and produce a uniform mixture. If alcohol, which is lighter than water, is poured on top of water, the heavier water will in the course of time rise and mix evenly with the alcohol. In the same way carbon dioxide, which is one and one-half times as heavy as air, will gradually

mix uniformly with the air. This property of matter is called diffusion.

MECHANICS.

Mechanics is that branch of physics which deals with forces in general and with their effects upon bodies. Mechanics is usually divided into *statics,* considering only such cases where the forces are in equilibrium, and in which no change of motion of the body acted upon takes place, and *dynamics,* which deals with those cases where a change in the motion of a body takes place under the influence of forces. Solids, liquids and gases are usually treated separately, and are treated in this book in the order named.

Forces.

By a force is meant that cause which, if a body is at rest, tends to put it in motion, and if in motion, produces, or tends to produce, a change in the velocity or direction of motion of the body. A push with the hand, the pressure of the wind, the expansion of steam are familiar examples of forces, that, if applied under suitable conditions, will cause changes in the motions of bodies or start motion from rest.

Graphic Representation of Forces.—Before the effect of a force can be examined, we must know first, the *point of application,* i. e., the point at which the force acts upon the body, secondly, its *direction,* and thirdly its *magnitude.* A force may very conveniently be represented by a straight line. Thus Figure 1 shows three forces, a, b and c, acting upon a body at the same point, in the directions shown by the arrows. The magnitudes of the forces are indicated by the lengths, using any convenient units.

Composition of Forces.—By composition of forces is understood the operation of finding one force which can replace two or more forces, this force being called the *resultant.*

Referring to Figure 1, it may readily be seen how the resultant of the two forces *a* and *b* acting at the same point is found. A line is drawn through the end of *a* parallel to *b,* another through the end of *b* parallel to *a,* and the intersection of these two lines is the end of the resultant, *ab.* This construction is called the *parallelogram of forces.*

If there are three forces, *a, b* and *c.* the resultant is found

as follows: In Figure 1, *ab*, as was just shown, is the equivalent of *a* and *b* in all respects. Therefore the resultant of the two forces *ab* and *c* will be the resultant of the three forces. The resultant *abc* is found by the parallelogram of forces, as shown in Figure 1, and this is in all respects the equivalent of *ab* and *c*, and consequently of *a*, *b*, and *c*. In this manner, the resultant of any number of forces acting at the same point of a body may be found.

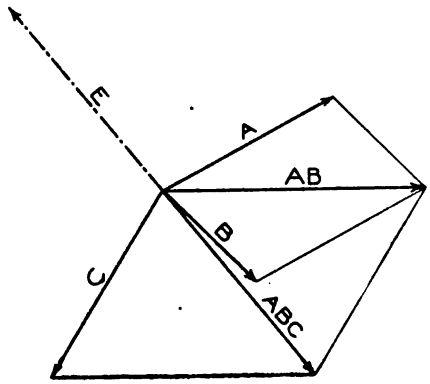


Figure 1. Graphic Representation of Forces.

The forces *a*, *b*, and *c*, shown in Figure 1, may be counterbalanced by a force *E* equal and opposite to the resultant *abc*. All the forces would then be in equilibrium.

If two forces act at the same point and in the same direction, the resultant is of course the sum of the forces. If two forces act at the same point and in opposite directions, the resultant is the difference between the two forces, and acts in the direction of the greater. If the two forces are equal and opposite, there will be equilibrium.

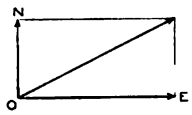


Figure 2. Graphic Representation of Forces.

Example.—If a body is pulled north by a force of 325 pounds, and east by a force of 650 pounds, acting at the same point, in what direction will the body move?

Solution.—(See Figure 2). The two forces are marked off, in this case using a scale of one-eighth of an inch equal to 100 pounds, as ON and OE. The parallelogram of forces gives OA as the resultant. In this direction therefore motion will take place.

Resolution of Forces.—By resolution of forces is understood the operation of replacing a single force by two or more forces, these forces being called *components.*

If, for instance, we have the force *abc,* Figure 1, acting at a point, it can be replaced by two component forces at the same point, in the directions indicated by *ab* and *c.* Also, if we have the force E acting at a point, two forces can be applied at the same point, in the directions indicated by *ab* and *c,* which will keep the force E in equilibrium. The process of constructing the components is just the reverse of that for constructing the resultant.

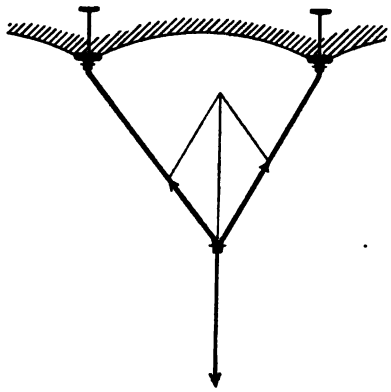


Figure 3. Graphic Representation of Forces.

Resolution of forces is especially useful for determining the reactions in the various kinds of supports. If, for instance, we tie a rope around a beam in the ceiling and suspend a certain body from it, the earth will exert a certain pull upon the body, and therefore upon the beam. The rope is in equilibrium and under a strong tension, and it is evident that there must be an opposite and equal pull upon the rope. Conditions are often such, that, instead of a single reaction force, two or more component forces are called into action.

For instance, in Figure 3, if a weight of 1,000 pounds were hanging vertically underneath one of the beams, there would simply be a pull of 1,000 pounds in the flat bar hanger. In the case shown in the Figure, however, the reaction force has to resolve itself into two components in the directions of the flat bars, and by using the parallelogram of forces the magnitude of these components may be found.

Moment of a Force.—The moment of a force with respect to a point is the product of that force and the perpendicular distance from the point to the force. Thus, if the axis of a body is fixed, though the body is free to rotate around the axis, the tendency to rotate does not depend alone on the size of the force acting upon the body, but also on the perpendicular distance between axis and force. This distance is called the arm of the force, and the force times the arm is equal to the moment of the force. The moment measures the tendency to produce rotation. In the case of pulleys and electric motors, the moment of a force is often called the *torque*.

If several forces are acting upon the same rotating body, the moments that tend to cause a clockwise rotation should be added together, and the moments that tend to cause a counter-clockwise rotation should be added together. The condition of equilibrium is that these two sums be equal. If the sum of clockwise moments is greater, a rotation in this direction will of course commence, and vice-versa.

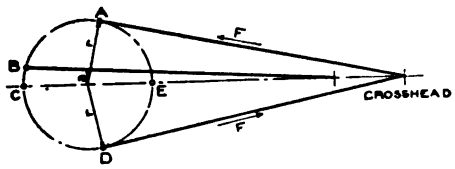


Figure 4. Crank Motion.

Examples of machinery parts acted upon by moments are met with in various parts of this book, for instance, the lever, the windlass, the gear wheel, the engine crank and many others.

In a crank motion (Figure 4) the length of the arm changes continually, reaching its greatest value r in the positions A and D of the crank pin (equal to half the stroke). In the position

B of the crank pin, the arm of the force is only the short length *a*, and in the positions C and E it is zero. If the connecting rod pushes with a constant force F for half of the revolution, and pulls with the same force for the other half revolution (which is only approximately true in practice), the moment of force is:

In the position A.....F x r
In the position B.....F x a
In the position C.....F x 0 = 0
In the position D.....F x r
In the position E.....F x 0 = 0

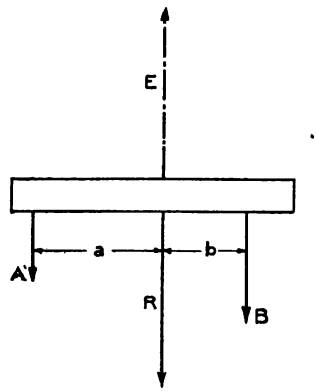


Figure 5. Parallel Forces.

Parallel Forces.—If parallel forces act upon a body, the resultant cannot be found by the method given for forces acting at the same point. In such cases, use must be made of moments of force. Thus, if the two parallel forces are A and B (Figure 5), they can be replaced by a resultant R, which is equal to A + B in size and acts upon the body at such a point, that the moments A x a and B x b are equal. The force E equal and opposite to force R will balance forces A and B.

Conversely, if we have the force R acting upon the body, and want to resolve it into two parallel components at the given points, the conditions are of course the same; i. e., A + B = R and A x a = B x b, only that it is now A and B that are the unknown quantities, while above it was *a* and *b* that were the unknown quantities.

Couples.—When two equal forces act at the same point in opposite directions, we have equilibrium. However, when the two equal forces in opposite directions act at different points on a body, i. e., the forces are not in the same straight line, but are parallel, we have what is called a couple. The perpendicular distance between the forces is called the arm of the couple (*a* in Figure 6). If we try to compute the resultant of a couple, we find that it cannot be replaced by a single force. A couple gives no motion of translation, only rotation.

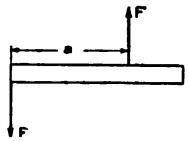


Figure 6. Couple.

Center of Gravity.—By center of gravity of a body is understood the point at which if suspended, there will be no tendency for rotation. It is the point through which the resultant of the parallel forces due to action of gravitation on the various particles passes. When we say that the earth is exerting a pull of, for instance, 10 pounds upon a body, it really means, that the earth is attracting every very small particle of the body with a proportionate very small force. All these forces are of course parallel, the magnitude of their resultant is 10 pounds, and the point through which the resultant passes in any position of the body, is called its center of gravity. If we suspend the body by a rope connected at this point, there must necessarily be equilibrium, as the reaction from the rope acts upon the same point as the resultant force of gravitation, with equal strength and in the opposite direction. In general it may be stated, that under any condition of influences from forces, a body will always act, as if the entire weight were concentrated in its center of gravity. It is well known that when we want to support a rod, of even thickness in the full length, at only one point, it must be at the center of gravity, i. e., exactly in the middle of the rod.

If the body is of uniform consistency, that is, if it is homogeneous, the location of the center of gravity depends only on the geometrical shape of the body.

All bodies generated by rotation—so-called bodies of revolution—have their center of gravity in the axis of rotation.

The center of gravity of a *ball* is its geometric center. The center of gravity of a *prism* lies in its axis in the middle of its length. The center of gravity of a *cylinder* lies in its axis of rotation in the middle of its height.

If each element of surface is regarded as a force or weight whose magnitude is proportional to its area, the center of gravity of such a *plane figure* is again the point where the resultant force, representing the total area, acts. The center of gravity of various plane figures is located as follows:

Parallelogram.—At the intersection of the diagonals.

Triangle.—At the intersection of the lines going from two angles to the middle of the opposite side.

Semicircle.—On the middle radius and 0.4244 x radius from the center.

Quadrant.—On the middle radius and 0.6002 x radius from center.

For figures of *irregular shape* it is advisable to have recourse to the following method: Draw the figure on a piece of paper of even thickness, cut it out with a knife and balance it over a sharp edge in two positions at right angles to each other. The intersection of the two marked traces from the edge on the paper is the center of gravity.

Trapezium or trapezoid.—Draw a diagonal, dividing the figure into two triangles. Draw a line through the centers of gravity of these two triangles. Draw the other diagonal and proceed in the same manner. The intersection of the two lines obtained is the center of gravity.

Equilibrium of Forces.—There are three kinds of equilibrium: stable, unstable and neutral equilibrium.

Figure 7 represents an iron disc in a vertical position with holes at the points A, B and C. If it is hanging on a horizontal pin through the hole at A, the disc is in *stable equilibrium,* and it is evident, that if swung out to either side, it will fall back to the position shown. This is because the center of gravity is below the point of suspension. If the disc is supported on a pin through the hole at B, it will stay in the position here shown,

if B is exactly below the center of gravity. But it is easily seen, that if the disc is swung a little out to one side, around B, it will not by itself return to the position shown, but fall until C is vertically below the pin. Therefore, when supported on a horizontal pin through B, the disc is said to be in an *unstable equilibrium.* When hanging on a pin through the hole at the center C (which is also the center of gravity), the disc will stay in any position; if swung to the side there is no tendency to change the position. Therefore the disc is said to be in *neutral equilibrium.*

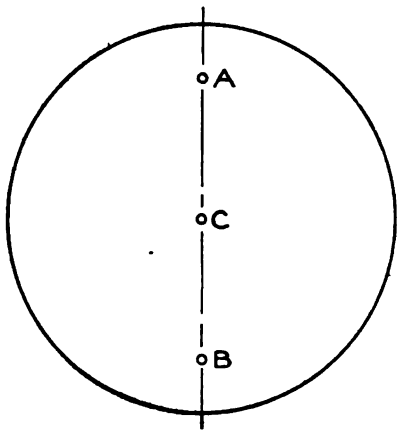


Figure 7. Equilibrium of Forces.

Motion of Bodies.

Velocity.—The path of a moving body may be either a straight line or a curve, and the motion may be uniform, accelerated or retarded. The rate of motion is commonly called *velocity.*

The velocity of a body at any instant is the distance it would pass over in the next unit of time without any influence from outside.

The distance traversed in a given time by a body moving with *uniform* velocity is equal to the velocity multiplied by the number of time units. If we call the distance s, the velocity v, and the time t, we have

$$s = v \times t;\ v = \frac{s}{t};\ t \times \frac{s}{v}$$

Examples.—If a train moves with a uniform velocity of forty (40) miles per hour, the *distance* traversed in three hours is $3 \times 40 = 120$ miles. If the train travels 120 miles in four hours, running at the same speed all the time, the uniform *velocity* is equal to $120 \div 4$, or 30 miles per hour. If the train moves 120 miles at a uniform velocity of 40 miles per hour, the *time* needed to traverse the distance is $120 \div 40$, or three hours.

Acceleration.—Acceleration is the change of velocity per unit of time, and may be either *positive* or *negative*. If the velocity increases, acceleration is positive; if the velocity diminishes, it is negative, and is termed *retardation.*

If a body has a uniformly accelerated motion and starts from rest, the velocity, after a certain time, is equal to the acceleration multiplied by the time. If, for instance, a body begins to move, and its speed increases seven feet per second, per second, the velocity, after three seconds, will be 3×7, or 21 feet per second. Representing the acceleration by a, the time by t, and the velocity by v,

$$v = a \times t.$$

The distance traversed if the body were moving at constant velocity would be $s = v \times t$, or $s = a \times t \times t$, but, since the velocity was zero at the beginning, the mean value of the velocity during the time t is

$$v = \frac{o + a\,t}{2} = \frac{a\,t}{2}$$

The distance traversed will then be

$$s = \frac{a \times t \times t}{2} = \frac{a\,t^2}{2}$$

If the initial velocity at the moment of starting of the acceleration were not zero, but uniform and equal to c, the velocity at the time t would be

$$v = c \pm a\,t.$$

The distance traveled would be

$$s = c\,t \pm \frac{a\,t^2}{2}$$

$+$ in case of acceleration.
$-$ in case of retardation.

Example.—A train, running at the rate of 60 miles per hour, has its speed reduced by brakes, at the rate of 2.93 feet per second, per second, making the time required for slowing down to standstill equal to 30 seconds. How far will the train run from start of braking to standstill?

As stated above, the distance is

$$s = c\,t - \frac{a\,t^2}{2}$$

The values for c, t and a are as follows:

$c = \frac{60 \times 5{,}280}{60 \times 60} = 88$ feet per second.

$t = 30$ seconds.

$a = 2.93$ feet per second.

Substituting these values in the formula, we obtain:

$$s = 88 \times 30 - \frac{2.93 \times 30 \times 30}{2} = 1{,}321.5 \text{ feet.}$$

Parallelogram of Velocities.—If a body moves under the influence of several forces, acting in various directions, its velocity will be a resultant of the velocities each force alone would produce.

Therefore a parallelogram of velocities can be constructed in the same way as a parallelogram of forces (see page 5).

Example 1.—A steamboat moves with a velocity of 10 miles per hour. What is the total or absolute velocity of a passenger walking on deck at a velocity of three miles per hour,

1) When going in the same direction as the boat?

2) When going in the opposite direction?

Solution.—1) The total or absolute velocity is the sum of the velocities of the boat and the passenger.

Velocity $= 10 + 3 = 13$ miles per hour.

2) The velocity is the difference.

Velocity $= 10 - 3 = 7$ miles per hour.

Example 2.—A ferry-boat is crossing a river at right angles at a velocity of 10 feet per second; the river itself flows at a velocity of 12 feet per second. What is the actual velocity of the boat?

Solution.—The velocities of the boat and river are represented, respectively, by the lengths OB and OF (see Figure 8).

They are at right angles to each other. OR is the resultant velocity, and is found as the hypotenuse of the right-angle triangle OBR or OFR, according to a well-known theorem:

$$OR^2 = OB^2 + BR^2 = 10^2 + 12^2 = 244,$$

whence $OR = \sqrt{244} = 15.62$ feet per second.

The direction in which the boat is going can be found by trigonometry.

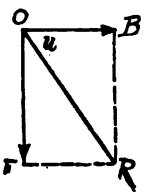


Figure 8. Paralellogram of Velocities.

Linear and Angular Velocity of Rotating Bodies.—By *linear velocity* of a rotating body is meant the feet per second any point in the body moves. By the *angular velocity* of a revolving body is meant the number of degrees through which any radius of the body moves in a second, or the rate in feet per second at which any point in the body having a radius of one foot moves.

The formulas are as follows:

$$\text{Linear velocity } V = \frac{2 \times 3.1416 \times r \times n}{60}$$

r = radius of body in feet.

n = number of revolutions per minute.

$$\text{Angular velocity } A = \frac{2 \times 3.1416 \times n}{60}$$

n = number of revolutions per minute.

This formula gives the angular velocity in so-called radians, one radian being $\frac{180}{3.1416}$ degrees. Hence, this result multiplied by $\frac{180}{3.1416}$ will give the angular velocity in degrees.

From this it is seen that the angular velocity equals the linear velocity divided by the radius.

Example.—A fly-wheel of 16 feet diameter makes 80 revolutions per minute. What are the linear and angular velocities of a particle on the circumference of the fly-wheel?

Solution.—$V = \frac{2 \times 3.1416 \times 8 \times 80}{60} = 67.02.$

The linear velocity is 67.02 feet per second.

$$A = \frac{2 \times 3.1416 \times 80}{60} \times \frac{180}{3.1416} = 480.$$

The angular velocity is 480 degrees per second.

Newton's First Law of Motion.—Every body persists in its state of rest or uniform motion in a straight line, unless it be compelled by some force to change that state. This property of a body is called *inertia.*

Gravitation.—Newton's law of gravitation states that every body in the universe attracts every other body with a force which varies inversely as the square of the distance between the two bodies, and is directly proportional to the product of the masses of the two bodies. In the case of the earth we have the attraction exerted by the earth on all bodies.

The distance is, in all cases, the *distance between the centers of gravity* of the bodies. If it were simply the open space between the two bodies, the force of gravitation would be infinitely great between the earth and bodies resting on the earth.

A body dropped from a place above the surface of the earth falls in a straight line and in the direction of the center of the earth, of course neglecting outside influences. Theoretically, all bodies, irrespective of weight, dropped from the same height would reach the earth at the same time. The resistance of the air, however, changes this, so that, for instance, a feather and a stone would not reach the earth at the same time. However, if the stone and feather were placed in a vacuum, a tube, for instance, from which the air is exhausted, and the tube inverted, both would fall with the same speed. Under such conditions, a body is said to *fall freely.*

In the following laws it is assumed that the bodies fall freely, which condition is approached more or less closely in air, according to the weight of the body. If a body starts from rest and falls freely, it will, at the end of the first second have a speed of 32.16 feet per second, that is, the acceleration in this case is 32.16 feet. This quantity is generally denoted by the letter g, and is a measure of the force of gravitation. At the equator g is

32.09 feet, at the pole 32.25 feet. Accordingly the velocity of a freely falling body is

$$v = g \times t,$$

that is, at the end of one second the velocity is 1×32.16 feet, or $1 \times g$; at the end of two seconds the velocity is 2×32.16, or $2 \times g$; after three seconds 3×32.16, or $3 \times g$, etc.

The space traversed is

$$s = \tfrac{1}{2} \times g\, t^2.$$

Hence in the first second the space traversed is $\tfrac{1}{2}\, g \times 1^2 = \tfrac{1}{2} \times 32.16 = 16.08$ feet; at the end of two seconds it is $\tfrac{1}{2}\, g \times 2^2$, after three seconds $\tfrac{1}{2}\, g \times 3^2$, etc. The space traversed is equal to 16.08 feet multiplied by the square of the number of seconds.

Example.—How deep is an artesian well if the report of a falling stone is heard in 5 seconds (disregarding time for sound)?

Solution.—Depth $= \dfrac{g\, t^2}{2} = \dfrac{32.16 \times 5 \times 5}{2} = 402$ feet.

From the above formulas can be deduced two other formulas of fundamental value in mechanical engineering, viz.:

$$h = \frac{v^2}{2g}$$

$$v = \sqrt{2g\, h.}$$

Example.—How high will a projectile rise vertically if it leaves the muzzle of a gun with a speed of 1,200 feet per second (disregarding resistance of air)?

Solution.—The height is

$$h = \frac{v^2}{2g} = \frac{1200^2}{2 \times 32.16} = 22{,}388 \text{ feet.}$$

Weight and Mass.—By *weight* of a body is meant the force of attraction of gravitation of the earth on the body, it being measured by a spring balance. As the force of gravity varies in different parts of the earth, so will the weight vary. In ordinary practice, however, weight means simply the amount of matter as determined by the ordinary scales compared with that of a standard unit of weight.

Mass = weight $\div$ g. Where the weight is determined on the common balance or platform scale, g is taken as 32.16, or approximately 32.2. Where the weight is determined by a spring balance, the value of g should be that of the locality in which the

weight is taken. In either case the value for mass of the body is constant in all parts of the world.

Newton's Second Law of Motion.—The rate of change of momentum of a body is proportional to the force acting on the body and is in the direction of that force.

The force neccessary to give a body a uniform acceleration is given by the following formula:

$$F = m \times a.$$

m = mass of the body.

a = acceleration.

Momentum.—The momentum of a body is dependent on the velocity and the mass, and may be defined as the product of mass and velocity ($m \times v$). If the velocity increases from v to $v + a$ during a second, the momentum has increased by the amount $m \times a$. Thus, as stated in Newton's Second Law, the force acting upon a body is measured by the change in momentum it produces per second.

Example.—An engine piston weighs 1,000 pounds, and moves in its central position with a speed of 600 feet a minute. What is the momentum of the piston at this speed?

Solution.—$\text{Mass} = \frac{1{,}000}{32.16} = 31.1.$

$$\text{Velocity} = \frac{600}{60} = 10 \text{ feet per second.}$$

$$\text{Momentum} = 31.1 \times 10 = 311.$$

Newton's Third Law of Motion.—For every action there is an equal and opposite reaction.

A familiar example of this is the recoil of a gun when a bullet is shot from it. The law finds frequent application to engineering problems.

Motion of a Projectile.—A body endowed with a certain velocity but without support will; under the influence of gravitation, describe a curved path similar to a parabola. The resistance of the air, however, which varies with the velocity of the body, alters to a great extent the results arrived at by elementary mechanics.

The water flowing through a horizontal opening from a tank also follows a parabola-like curve.

The Pendulum.

A simple pendulum (Figure 9) consists of a ball suspended by a string which is so light as compared to the ball, that its weight may be disregarded. If the ball is moved a distance c to the left, it is necessarily lifted a height h at the same time. If it is now let loose and allowed to move without interference, the force of gravity will pull it downward, it will swing towards the lowest point, and its inertia will carry it by this point. As the ball travels towards the right, it is compelled to rise; the force of gravity opposes the rise and decreases its momentum. If it were not for the resistance of the air, the ball would not stop until it had reached the same distance c to the right, and thus it would continue, swinging equally far on each side. However, the air resistance causes the distance that the ball travels away from the vertical center line to be shorter and shorter each time, until finally it comes to rest in the center position.

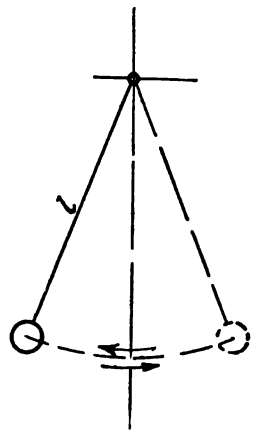


Figure 9. Pendulum.

The interesting feature of the pendulum motion is, that the swinging from the extreme position on the left side to the extreme position on the right side, requires the same time, whether the distance c is great or small. This length of time is called the *vibration time*. It is also independent of the weight of the ball, but is proportional to the square root of the length of the pen-

dulum. The formula for the vibration time of a simple pendulum is:

$$t = 3.1416\sqrt{\frac{l}{g}}$$

t = vibration time in seconds.
l = length of the pendulum in feet.
g = force of gravity = 32.16 at this latitude.

From this formula the value of g at different places of the globe may be found.

A body of any shape suspended from a fixed axis about which it oscillates by the force of gravity, is called a *compound pendulum*. As with the simple pendulum it has a fixed time of vibration, but the formula for this is rather complicated.

Example.—A simple pendulum is made up of a ten-pound weight suspended by an ordinary string, the distance from the point of suspension to the center of the weight being 8 feet. What is the time of vibration?

Solution.—$t = 3.1416\sqrt{\frac{8}{32.16}} = 1\frac{1}{2}$ seconds approximately.

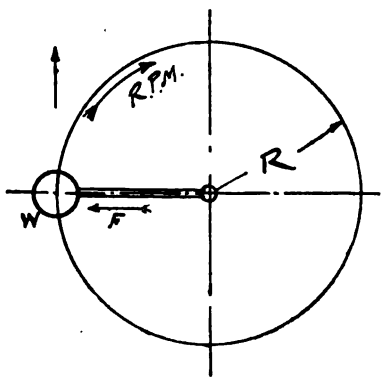


Figure 10. Centrifugal Force.

Centrifugal Force.

A body W, fixed at one end of a solid rod or cord and forced to revolve about an axis, exerts upon this rod or cord, a force F radially outward, which is called centrifugal force (see Figure 10). The formula for this force is as follows:

$$F = \frac{W \times R \times (R.P.M.)^2}{2{,}934}$$

F = centrifugal force in pounds.
W = weight in pounds.
R = radius in feet.
R P.M. = revolutions per minute.

Example.—What is the centrifugal force produced by a ball of 10 pounds weight revolving at a radius of 1.2 feet and making 240 revolutions per minute?

Solution.—$F = \frac{10 \times 1.2 \times 240 \times 240}{2{,}934} = 235.6$ pounds.

In case of rupture of the radial support the ball flies off at a tangent with a velocity of:

$$v = \frac{2 R \pi n}{60} = \frac{2 \times 1.2 \times 3.1416 \times 240}{60} = 30.16 \text{ feet per second.}$$

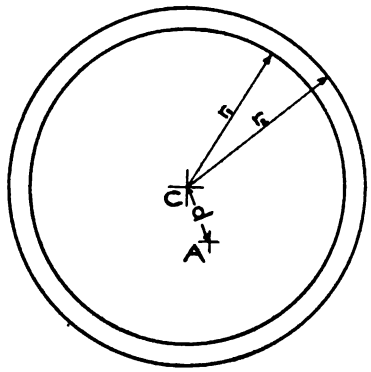


Figure 11. Moment of Inertia.

Moment of Inertia.

The moment of inertia of a body is calculated as follows: The body is divided into a number of small parts, the more the better. The mass of each small part is multiplied by the square of its distance from the axis of rotation, and all the products are added together. The moment of inertia is not a fixed quantity for any body, but changes in the same body when another axis of rotation is used. For a vertical ring rotating around a horizontal axis through its center (see Figure 11), the moment of inertia is:

$$I = M \times \frac{r_1^2 \times r_2^2}{2}$$

M = total mass of the ring.

r_1 = radius of inside of ring.

r_2 = radius of outside of ring.

If the axis instead of being in the center is located at A, at a distance d from the center, the moment of inertia becomes:

$$I = M \times \frac{r_1^2 \times r_2^2}{2} + M \times d^2.$$

In calculations where the moment of inertia enters, it is often convenient to have it in this form:

$$I = M \times k^2.$$

k is a distance of such length, that when squared and multiplied by the mass it gives the moment of inertia of the body. The length k is called the *radius of gyration.*

Work, Energy and Power.

Work.—In general, when a force causes motion of a body through space, it is said to do work on that body. The unit of work is the amount of work required to raise one pound one foot against the force of gravity, and is called a *foot-pound.* Three pounds raised 10 feet, or 10 pounds raised three feet, represent 30 foot-pounds. Where the metric system is used, the unit of work is the meter-kilogram.

The work required to move a body through space is given by the following formula:

$$E = f \times s.$$

E = work required.

f = force or resistance overcome.

s = distance through which force acts.

In the case of gravity, we have

$$E = w \times h.$$

E = work required.

w = weight of body.

h = vertical distance body must be lifted.

Energy.—The energy of a body is its capacity for doing work, and is measured in the same units as work.

Kinetic and Potential Energy.—A moving body can do work by imparting motion to other bodies, and consequently a

moving body possesses energy. The energy of a moving body is called *kinetic energy,* and is given by the following formula:

$$E = \frac{m v^2}{2}$$

E = energy in foot-pounds.
m = mass of body.
v = velocity of body in feet per second.

A body at rest may also have energy, if it is in such a position that it will move if allowed to do so. Work has been done on such bodies and energy is stored up in them. Such energy which can be released and do work is called *potential energy.* Potential energy is stored up, for instance, in a lifted weight, in a stretched or compressed spiral spring, in compressed air, etc. If a stone is thrown vertically upward, the kinetic energy imparted to the stone does the work of lifting the stone against the force of gravity. When the stone reaches the highest point and comes to rest, the kinetic energy has all been spent and transformed into as many foot-pounds of work as are represented by the weight of the stone in pounds multiplied by the number of feet it has been raised. The kinetic energy has been changed into so many foot-pounds of potential energy, which, during the fall of the stone, again reappears as kinetic energy.

The energy as given above is equal to the potential energy necessary to produce the velocity irrespective of the time, according to the law of the conservation of energy.

$$E = W \times h = \frac{m v^2}{2} = \frac{W}{g} \times \frac{v^2}{2}$$

Hence $h = \frac{v^2}{2g}$, or $v = \sqrt{2gh}$.

Example.—What velocity and kinetic energy are attained by a body dropping a height of 64 feet, if the body weighs 200 pounds?

Solution.—The velocity does not depend upon the weight.
$v = \sqrt{2 g h} = \sqrt{2 \times 32.16 \times 64} = 8.02 \times \sqrt{64} = 64.16$ feet per second.

The mass $= \frac{200}{32.16} = 6.219$, hence the kinetic energy attained is

$$E = \frac{m v^2}{2} = \frac{6.219 \times 64.16^2}{2} = 12{,}800 \text{ foot-pounds.}$$

The potential energy before falling is

$$E = W \times h = 200 \times 64 = 12{,}800 \text{ foot-pounds.}$$

Conservation of Energy.—The fundamental law upon which all our machinery is built up, the law of conservation of energy, is expressed in this way: *Energy can be transformed, but cannot be created or destroyed.*

As a good example of the conservation of energy, we may take the case of a steam engine. A pound of coal containing energy in form of heat of about 12,000 B.T.U. will impart to the water in a well-designed boiler only about 8,000 B.T.U. Of the remaining 4,000 B.T.U. the larger part is carried out through the chimney by the escaping gases; the smaller represents the loss of heat due to radiation, convection and conduction. The greatest part of the 8,000 B.T.U. is required to form steam from water, which heat either passes into the free exhaust when the steam engine is non-condensing, or is taken up by the cooling water when the engine is condensing. About 1,500 B.T.U. are available at the engine, but only a certain per cent of this is turned into useful work, on account of losses due to friction, condensation, etc. The remaining energy serves to drive the desired apparatus or machine, in which friction and other resistances transform the energy into heat, which is dissipated into space; consequently all the energy in the coal has returned back to nature, none being lost. For the particular purpose of the machine, however, only a small amount of this energy can be utilized.

Power.—Power is the time-rate of doing work. The unit of power is that power which can do the unit of work (1 foot-pound) in the unit of time (1 second).

Horse-power.—For measuring the power of engines, etc., a unit called a horse-power is used, which represents work at the rate of 33,000 foot-pounds per minute, or $33{,}000 \div 60 = 550$ foot-pounds per second. The formula for horse-power is as follows:

$$\text{Horse-power} = \frac{w \times v}{33{,}000}$$

w = weight in pounds lifted.

v = velocity in feet per minute.

Below are given two examples on horse-power. More details on this subject may be found under "Steam Engines" and "Pumps." These are merely to give a clear conception of the meaning of the term.

Example 1.—How many horse-power are theoretically necessary to raise a weight of 1,000 pounds at a velocity of 50 feet per minute?

Solution.—Horse-power $= \frac{1{,}000 \times 50}{33{,}000} = \frac{50{,}000}{33{,}000} = 1.5.$

Example 2.—What is the theoretical output of a water turbine working under a head of 15 feet with 4,200 cubic feet of water per minute?

Solution.—One cubic foot of water weighs approximately 62.4 pounds, therefore 4,200 cubic feet weigh $4{,}200 \times 62.4 = 262{,}080$ pounds.

$$\text{Horse-power} = \frac{262{,}080 \times 15}{33{,}000} = 119.2.$$

Impact of Bodies.

If two inelastic bodies, which have different velocities and travel in the same direction, collide, they will move on as one mass with a velocity:

$$C = \frac{m \times v + M \times V}{m + M}$$

M and m = masses of the two bodies.

V and v = velocities of the two bodies before impact.

C = velocity after collision.

Friction.

If one body lay over another and each had a perfectly smooth surface, the least exertion of a force would cause one to slide over the other. However, there is no possibility of obtaining such surfaces. All surfaces which we can produce are rough when examined under the microscope, and exhibit projections and recesses which resist the sliding of one body over the other. This resistance is called friction.

Sliding Friction.—The resistance offered by one body

sliding over the other is called the *frictional resistance.* The surface of contact is called the *frictional or rubbing surface.* The ratio of the friction to the force creating it is called *coefficient of friction.*

1. Frictional resistance increases with the pressure.

2. Frictional resistance is independent of the area of the frictional surface.

3. Frictional resistance depends on the condition and nature of the frictional surfaces. Smooth and hard surfaces have less frictional resistance than rough and soft surfaces, and making the surfaces smooth by using lubricants, will reduce the frictional resistance.

4. Frictional resistance when starting to move is greater than when in motion; we therefore speak about *friction of rest* and *friction of motion.*

5. Frictional resistance while the body is in motion is not quite independent of the velocity of the moving body.

6. Pivot frictional resistance is less than sliding frictional resistance.

The formula for frictional resistance is:

$$F = m \times N.$$

F = frictional resistance.

m = coefficient of friction.

N = pressure producing friction.

Rolling Friction.—When a cylindrical body rolls on a plane surface it overcomes the so-called rolling friction, which is very much smaller than the sliding friction. It can be determined by experiment, the cylinder being placed upon an inclined plane whose slope is adjustable. It is found that the resistance due to rolling friction increases with the pressure, but that it decreases when the roller diameter is increased. It is larger the softer the material is, which influence enlarges the arc of contact.

The formula for rolling friction is:

$$R = \frac{w \times f}{r}$$

R = resistance of rolling.

w = weight in pounds.

f = coefficient of rolling friction.

r = radius of wheel in feet.

The value of the coefficient f varies considerably. The following are some values, which may be taken as approximate:

Iron against iron .....f = 0.002
Hard wood against wood.....f = 0.02
Soft wood against wood.....f = 0.04
Stone against stone .....f = 0.06

Table of Friction Coefficients.

Description of the Bodies under Friction		Dry	Moist	Olive Oil	Pig Lard	Tallow	Pig Lard with Graphite	Refined Wagon Lubricant	Dry Soap
Wood against wood	Minimum value	0.20	....	....	0.06	....	....	....	....
	Mean value ...	0.36	0.25	....	0.07	0.06	....	....	0.14
	Maximum value	0.48	....	....	0.07	0.07	....	....	0.14
Metal against metal	Minimum value	0.15	....	0.06	0.07	0.08	....	....	0.16
	Mean value ...	0.18	0.31	0.07	0.09	0.07	0.06	0.12	....
	Maximum value	0.24	....	0.08	0.11	0.09	0.08	0.15	0.20
Metal against wood or reversed	Minimum value	0.20	....	0.05	0.07	0.11	0.09	0.17	....
	Mean value ...	0.42	0.24	0.06	0.07	0.06	....	....	....
	Maximum value	0.62	....	0.08	0.08	0.08	0.08	0.10	0.20
Manila rope against	Wood	0.45	0.33	....	....	0.10	....	....	....
	Metal	....	....	0.15	....	....	....	....	....
Sole leather with surface against wood or metal...	Rough	0.54	0.36	0.16	....	0.19	....	....	....
	Hammered	0.30	....	....	....	0.20	....	....	....
Sole leather with edge against wood or metal	Dry	0.34	0.31	0.14	....	....	....	....	....
	Lubricated	....	0.24	....	....	....	....	....	....
Dry pulverized soil...		0.94	....	....	....	....	....	....	....
Moistened soil		....	1.38	....	....	....	....	....	....
Fine dry sand.....		0.69	....	....	....	....	....	....	....

ELEMENTS OF MACHINES.

The elementary machines may be classified thus:

(1) The Lever.
(2) The Pulley.
(3) The Inclined Plane.

In the following description, frictional resistance is disregarded, the work spent on a machine being considered as equal to the work obtained.

The Lever.

The lever is a solid body, commonly a bar, which is supposed to be not flexible. It has a fulcrum, or fixed axis, (Figures 12 to 15), and at least two forces acting upon itself. The force which resists the motion of the lever is the *load* L. The force which moves the lever is called the *force* F. For general calculations the weight of the lever itself is neglected.

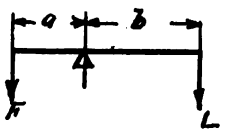


Figure 12. Lever, First Class.

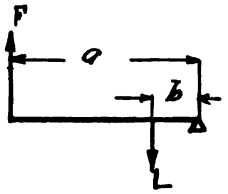


Figure 13. Lever, Second Class.

Levers are generally divided into three classes, as follows:

In the first class, the fulcrum lies between the force and the load (Figure 12). Examples of this class are the scissors and the crow-bar.

In the second class, the load is between the force and the fulcrum (Figure 13). Examples of this class are a nut-cracker, and a claw-hammer for extracting nails.

In the third class, the force acts between the fulcrum and the load (Figure 14). Examples of this class are the tongs, and the lever of a safety-valve.

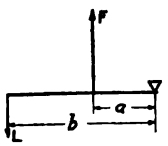


Figure 14. Lever, Third Class.

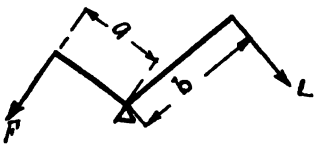


Figure 15. Angular Lever.

When the slope of the lever is such that the acting forces form angles of about 90°, the lever is called a *bent lever, bell crank* or *angular lever* (Figure 15).

Force and load on a lever are in equilibrium when the moments of force (see page 7) with respect to the fulcrum are

equal. This means that in all above cases the following equation must be fulfilled:

$$aF = bL.$$

$$F = \frac{b}{a} L.$$

Example.—How great is force F, if a = 6 feet, b = 2 feet, and L = 27 pounds?

Solution.—$F = \frac{2}{6} \times 27 = 9$ pounds.

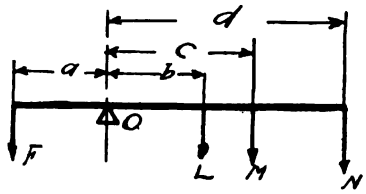


Figure 16. Lever, Several Forces Acting.

If more than two forces are acting (Figure 16), the equation reads:

$$aF = bL + cM + dN.$$

$$F = \frac{bL + cM + dN}{a}$$

Example.—How great has force F to be if the following values are assumed: a = 10 ft., b = 5 ft., c = 12 ft., d = 22 ft., L = 40 lbs., M = 30 lbs., N = 5 lbs.?

Solution.—$F = \frac{5 \times 40 + 12 \times 30 + 22 \times 5}{10} = 67$ pounds.

The total pressure upon the edge O equals 40 + 30 + 5 + 67 = 142 pounds.

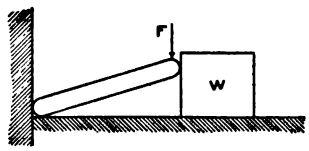


Figure 17. Moving Strut.

Moving Strut.—The moving strut is shown in Figure 17. It is similar to a bent lever, except that force and load act at the same end of a single arm.

Toggle-joint.—The toggle-joint consists of two moving struts placed end to end, the force being applied at the junction, the other end of one of the struts resting against a fixed abutment, and that of the other against the body to be moved. This device serves to exert very large forces with a very small displacement of the load, and finds application in stonecrushers.

The Windlass.—The windlass or wheel and axle (Figure 18) may be called a perpetual lever, as it consists of two wheels on one axis, having different radii, which can be considered as lever arms.

The moments of load and force are;

$$F \times R = L \times r.$$

$$F = \frac{r \times L}{R}$$

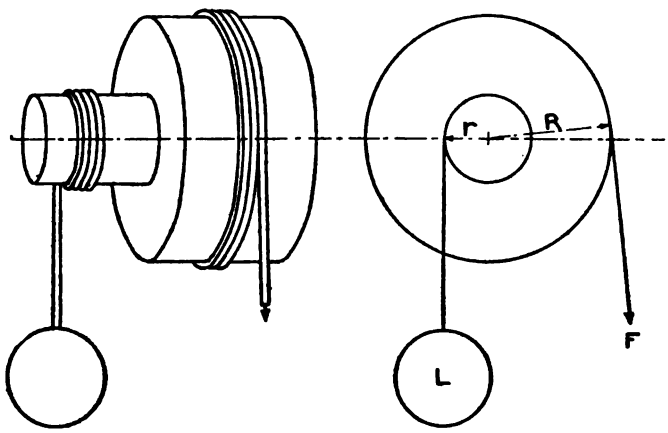


Figure 18. Wheel and Hoisting Drum.

Toothed-wheel Gearing.—This consists of several groups of wheels and axles. Such combinations are quite frequently used as hoisting machines (see Figure 19).

The force F required to lift the load L is found as follows:

In group A, $L \times t = G \times T$, whence $G = \frac{t}{T} \times L$.

In group B, $G \times s = H \times S$.

Inserting value for G,

$$\frac{t}{T} \times L \times s = H \times S, \text{ whence}$$

$$H = \frac{t}{T} \times \frac{s}{S} \times L.$$

In group C, $H \times r = F \times R$.

Inserting value for H,

$$\frac{t}{T} \times \frac{s}{S} \times L \times r = F \times R, \text{ whence}$$

$$F = \frac{t}{T} \times \frac{s}{S} \times \frac{r}{R} \times L.$$

$\frac{t}{T}, \frac{s}{S}, \frac{r}{R}$ are *lever ratios,* also called *transmission ratios,* and are found in the case of gears as the ratio of the number of teeth of the two wheels on the same shaft.

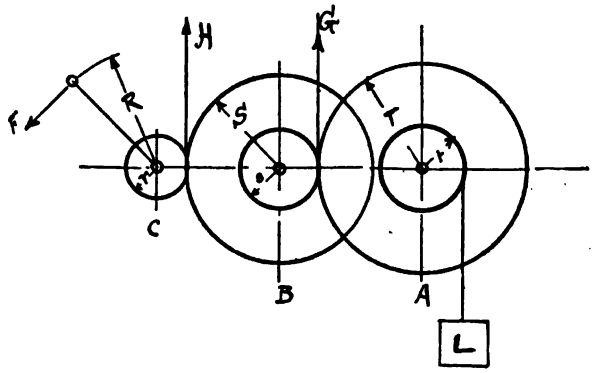


Figure 19. Toothed Wheel Gearing.

If v denotes the velocity of the load L, and C the velocity or the force F, then, since the work per second is equal for both,

$$F \times C = L \times v$$

$$C = \frac{L}{F} \times v.$$

$$v = \frac{F}{L} \times C.$$

Nothing has been gained in power, as the force F has to travel faster than the load L in the same ratio as the load has been decreased.

Example.—Referring to Figure 19, assume $L = 4{,}700$ pounds, $t = 6$ inches, $R = 18$ inches.

Number of teeth of wheel A.....120

Number of teeth of small wheel B..... 24

Number of teeth of large wheel B..... 72
Number of teeth of wheel C..... 10

How large is the necessary force F and how fast will the load L rise, if the crank handle is moved at a velocity of 2.5 feet per second?

Solution.—$F = \frac{24}{72} \times \frac{6''}{18''} \times \frac{10}{120} \times 4{,}700 = 43.52$ pounds.

(This would require two men at the crankhandle, as no friction has been taken into account).

The load rises at a velocity of

$$v = \frac{43.52}{4{,}700} \times 30'' = 0.2778 \text{ inch per second.}$$

Figure 20.
Simple Pulley.

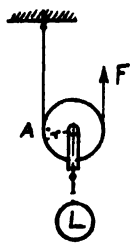


Figure 21.
Movable Block.

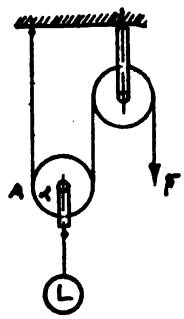


Figure 22.
Movable Block.

Pulleys or Blocks.

The pulley consists of a narrow disc turning around its axis with a rope going over the grooved circumference. There are stationary blocks, movable blocks, multiple blocks as a combination of both, and differential blocks.

In the stationary block, Figure 20, load and force are equal; the whole advantage lies in the change of direction of the force F.

In the movable block, Figure 21, the work done upon the pulley by the force equals the work obtained from the pulley. Lifting L one foot requires pulling of F through two feet:

$$2 \times F = 1 \times L$$
$$F = \tfrac{1}{2} L$$

The force F is thus only half of the load, while its velocity must be twice the velocity of the load.

Figure 22 shows the movable block as it is used in ordinary practice.

Figure 25 shows a tackle of pulleys very much in use.

The load L, is carried by six ropes, each of which is pulled with force F, hence:

$$L = 6\,F, \text{ or in general} = 2\,n\,F.$$

n = the number of loose pulleys.

The velocity of the load is

$$v = \frac{C}{2n}$$

C = velocity of force.

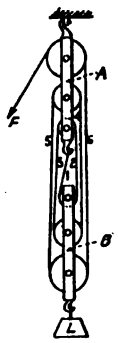


Figure 23.
Block and Fall.

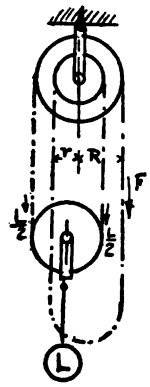


Figure 24.
Differential Pulley.

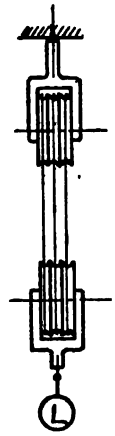


Figure 25.
Triple Movable Pulley.

The *differential pulley* consists of two fixed rollers of different diameters cast together on the same axis, and an ordinary loose pulley connected with an endless chain (see Figure 24).

The chain runs off from the large pulley through force F and runs over the smaller one to the loose pulley and back to the large one.

With respect to C as a fulcrum, we have a double-armed lever, and therefore

$$F \times R + \frac{L}{2} \times r = \frac{L}{2} R,$$

$$F = \frac{L}{2}\left(1 - \frac{r}{R}\right)$$

$$V = \frac{C}{2}\left(1 - \frac{r}{R}\right)$$

If R and r are nearly alike, the device will be self-locking. The usual proportion in differential pulleys on the market is

$$\frac{r}{R} = \frac{11}{12}$$

which gives a velocity ratio of:

$$\frac{1}{2} \times \left(1 - \frac{11}{12}\right) = \frac{1}{24}$$

Example 1.—How great a force is necessary to lift 600 pounds with a loose pulley weighing 10 pounds?

Solution.—$F = \frac{L + \text{pulley}}{2} = \frac{600 + 10}{2} = 305$ pounds.

Example 2.—What load can be lifted with a differential pulley with a ratio $\frac{r}{R} = \frac{11}{12}$, the force being 100 pounds?

Solution.—$L = \frac{2F}{1 - \frac{r}{R}} = \frac{200}{1 - \frac{11}{12}} = 200 \times 12 = 2{,}400$ pounds.

The *Differential Windlass* is similar in principle to the differential pulley, but is seldom used on account of the length of rope required.

The Inclined Plane.

If a body rests on an inclined plane, by resolution of forces, we find that in all respects the effect of the forces F and N are equivalent to the effect of W, the force with which the earth strives to pull the body vertically down. N calls forth an equal and opposite reaction from the plane, but cannot possibly give any motion, while F cannot possibly cause any reaction, but will immediately begin to create velocity, if not counteracted. We may therefore state, that under these circumstances the effects of the gravitation force W are, to exert a pressure N perpendicular to the inclined plane, and to pull the body along the plane with a force F. The condition of equilibrium is to apply a force equal to F in magnitude and in the opposite direction.

Three cases may arise here. The sustaining force F may act parallel to the plane, parallel to the base, or at angle to either

plane or base. In the first case (see Figure 26), the force required to sustain the load is found as follows:

$$F = W \times \frac{CB}{AC}$$

W = weight of body.
CB = height of plane.
AC = length of plane.

In the second case, where the force acts parallel to the base, the force is

$$F = W \times \frac{CB}{AB}$$

CB = height of plane.
AB = length of base of plane.

$\frac{CB}{AB}$ is called the grade or slope, and is usually given in per cent. Thus, a rise of 1 per cent means a rise of 1 per cent of 5,280, or 52.8 feet per mile.

Where the grade is very small, it is not important to specify the exact direction of the force. It may be horizontal or parallel to the plane without changing the result appreciably.

In the third case, where the force acts at an angle, the force varies with the position of W, and hence no definite rule can be given.

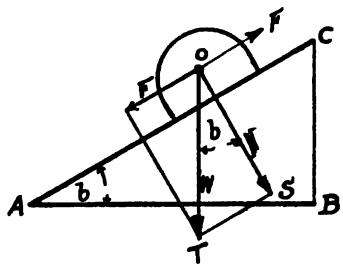


Figure 26. Inclined Plane.

Example.—The length of an inclined plane is 12 feet, and its height 2 feet. What force is required, acting parallel to the plane, to sustain a weight of 180 pounds?

Solution.—$F = W \times \frac{CB}{AC} = 180 \times \frac{2}{12} = 30$ pounds.

The Wedge.—A body shaped such that two of its main surfaces form a knife edge is called a wedge. The two planes can

be considered as inclined planes, therefore, the forces have to be resolved in the same manner (see Figure 27).

Triangles ABC and SVO being similar, the following proportion can be set up:

$$OV : OS = AB : AC$$
$$F : L = AB : AC$$
$$OV \text{ being } = F, \; OS \text{ being } = L.$$
$$F = L \times \frac{AB}{AC}$$

the force applied on the head of the wedge.

Example.—What force will produce a wedging action of 1,000 pounds if $AB = 1''$, $AC = 10''$?

Solution.—From the above formula,

$$F = 1{,}000 \times \frac{1}{10} = 100 \text{ pounds.}$$

Friction is entirely disregarded, as in all the fore-going examples.

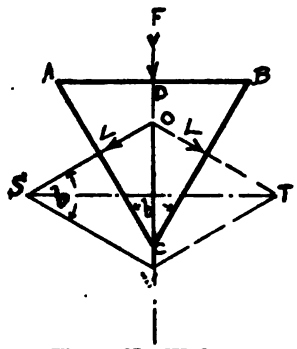


Figure 27. Wedge.

The Screw.—The screw represents an inclined plane wound over the surface of a cylinder, the height of the plane being parallel to the axis of the cylinder. If on the internal surface of a hollow cylinder, it is called a nut. When used for lifting a weight or overcoming a resistance, the force is usually applied by means of a wrench or wheel. The force is independent of the diameter of the screw.

Neglecting friction, the force required to overcome a certain resistance may be expressed by the following formula:

$$F = \frac{W \times p}{2 \times 3.1416 \times r}$$

F = force applied in pounds.

W = weight lifted or resistance overcome in pounds.

p = pitch of screw or distance in inches traveled in one revolution of the screw.

r = radius of wheel or the length of the lever-arm in inches.

Example.—If the pitch of a screw is ½ inch, what force is necessary to produce a pressure of 10,000 pounds, the force acting on a lever, the lever-arm being 20 inches. Friction is neglected.

Solution.—$F = \frac{10{,}000 \times 0.5}{2 \times 3.1416 \times 20} = 39.79$ pounds.

The greatest application of the screw is found in the design of machines, where distinction is made between *fastening screws* and *transmission screws.* The former kind uses the considerable friction produced in the thread and on the seat of the nut; it is *self-locking.* In the latter kind either the nut or the spindle is turned, whereby a rectilinear motion of one of them is produced. Only with very great pitch which necessitates several independent threads running in parallel is it possible to produce a rotary motion with a rectilinear movement.

Cam.—The cam is a revolving inclined plane. The calculation is the same as that given for the screw.

Worm-gear.—The worm-gear works on the same principle as the screw, and is mostly used for considerable speed reductions (see also page 439).

Efficiency of Machines.

If a machine is designed to produce power, it is necessary that the primary source procure an amount of energy available for the machine which is larger than the output of the machine. This means that the input must under all circumstances be larger than the output. This is due to the fact that in all changes from one form of energy into another a certain part of the energy spent is useless work, mostly in overcoming friction. The energy thus expended is transformed into heat which is dissipated into the atmosphere.

Let W = useful work furnished by machine.

L = losses.

U = total input of energy.

E = efficiency of machine.

Then $U = W + L$.

$$E = \frac{W}{U} = \frac{W}{W + L}$$

Thus it is seen that the efficiency is always less than unity, and therefore a mechanism which would perform an amount of work greater than that put into the machine is an impossibility.

STATICS OF LIQUIDS.

Liquids are practically incompressible; even very heavy pressure fails to reduce their volume by any appreciable amount.

Pascal's Law.

If a liquid is enclosed in a vessel and a pressure is exerted upon a part of its surface, that pressure is transmitted undiminished in all directions, and acts with the same force upon all surfaces, and in a direction at right angles to those surfaces.

This law finds an application in the *hydraulic press,* which consists of two cylinders of unequal diameters, communicating with each other and filled with water. A pressure downward on the water in the narrow cylinder produces a pressure upward in the wide cylinder, as many times greater, as the sectional area of the wide cylinder is greater than that of the narrow cylinder.

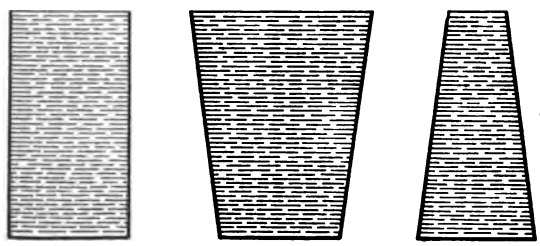


Figure 28. Downward Pressure in Liquids.

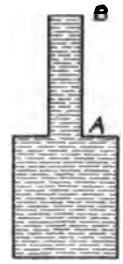


Figure 29. Upward Pressure in Liquids.

Downward Pressure.—If a liquid is contained in a vessel, the pressure on the bottom is independent of the shape of the vessel, and depends only on the depth of the liquid and the area of the bottom. If the three vessels shown in Figure 28 are filled with water and each has the same depth and area of bottom, the pressure on the bottom of each vessel is the same, even though the second vessel contains considerably more, and the third con-

siderably less, water than the straight cylindrical vessel. When there is no pressure at the surface, this may be expressed in the following law:

The total pressure on the bottom of a vessel containing a liquid is independent of the shape of the vessel and is equal to the weight of a column of the liquid, the area of whose base is equal to the area of the bottom of the vessel, and whose height is equal to the depth of the bottom below the surface of the liquid.

Example.—Referring to the vessels in Figure 28, assume them to contain water having a depth of 3 feet, and the area of the base of each to be 1 square foot. What is the pressure in pounds on the bottom of each vessel?

Solution.—According to the law just given, the pressure in each case must be figured on a column of water 3 feet high with a cross-sectional area of 1 square foot. The pressure of such a column will be $3 \times 1 \times 62.4 = 187.2$ pounds, this consequently being the total pressure on the bottom in each case. The pressure in pounds per square inch would be $\frac{187.2}{12 \times 12} = 1.3$ pounds.

Upward Pressure.—In the vessel, shown in Figure 29, which is filled with water, there is an upward pressure on b caused by the small column of water ab. The amount of this pressure is equal to the weight of column of water whose height is equal to ab and whose cross-sectional area equals the area of b.

Example.—A horizontal surface, whose area is 3 square feet is submerged in a vessel of water at a depth of 2 feet below the upper surface. What is the total upward pressure on the submerged surface?

Solution.—According to the rule given, the weight of the equivalent column of water is $3 \times 2 \times 62.4 = 374.4$ pounds, which is the total upward pressure on the submerged surface.

Lateral Pressure.—Referring to Figure 28, there is, on the vertical sides of the first vessel at a point one foot below the water level, a pressure of 0.433 pound on one square inch or area and at right angles to the side. At a point two feet below the water level there is a pressure of $2 \times 0.433 = 0.866$ pound on one square inch of area. In the second and third vessels there are the same pressures on the sides at points at the same depths, the only difference being that the directions of the

pressures are different, in each case perpendicular to the side. This may be summed up in the following law:

The pressure on any surface below a liquid is equal to the weight of a vertical column of the liquid, whose base has an area equal to that of the said surface, and whose height is the depth from the level of the liquid to the center of gravity of the surface.

Example.—A rectangular vessel is filled with water. What is the total lateral pressure on a part of the wall two feet wide and 1 foot deep, the top of which part is 7 feet below the surface of the liquid.

Solution.—Since the top of the part of the wall under consideration is 7 feet below the surface, and bottom $7 + 1 = 8$ feet below, the center of this part is $\frac{7+8}{2} = 7.5$ feet below the surface. The area is $2 \times 1 = 2$ square feet. Hence the total pressure is $7.5 \times 2 \times 62.4 = 936$ pounds.

Liquids in Connecting Vessels.

From the above laws it follows that the water level in connecting vessels, no matter what their shape may be, is the same.

If the water level were higher in any one of the vessels than

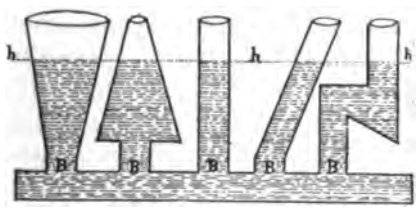


Figure 30. Liquids in Connecting Vessels.

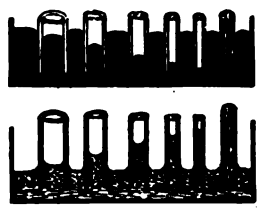


Figure 31. Capillary Phenomena.

in the others, it would follow that the downward pressure of the water in this vessel would be greater than that of the others. Consequently there would not be equilibrium, and motion of the water would take place until the liquid in all the vessels had the same level (see Figure 30).

A good example of this principle is shown in the gauge glass attached to a steam boiler.

Surface Tension.

The surface of a liquid is physically different from the interior. It is in a state of tension similar to that of a membrane, like a

drumhead stretched equally in all directions. A good example of surface tension is the soap bubble. Surface tension of a liquid has the tendency to have the liquid assume the smallest possible surface for a given volume, for which reason water-drops or drops of molten metal assume the spherical form.

It is due to surface tension that it is possible to float a polished steel needle upon water, and that some forms of water bugs move on the surface, although they are heavier than the water.

Water, next to mercury, has the greatest surface tension of all liquids. When a drop of oil is placed upon water it immediately spreads over the surface, this being due to the stronger surface tension of the water, which tension, however, is thereby weakened.

Capillary Phenomena.

If we place a clean, narrow glass tube, open at both ends, vertically down in a liquid, the level of the liquid is usually either higher or lower than the surface of the liquid in the vessel. In the case of water the liquid is higher, in the case of mercury lower (see Figure 31). It will also be noticed that the surface of the liquid in the tube is curved, in the case of water upwards and in the mercury downwards. The elevation or depression of the liquid depends on the diameter of the tube, being greater the smaller the tube. These conditions are called capillary phenomena. The deviations in height are always small and they occur only in comparatively narrow tubes, not in tanks or wide pipes.

Archimedes' Principle.

If a body is immersed in a liquid it displaces an equal volume of the liquid. At the same time the apparent weight of the body in the liquid is less than its true weight in air. The *loss of weight* is equal to the weight of the displaced liquid. This is called the principle of Archimedes, it having been discovered by him. A cubic foot of stone weighs, for instance, 169 pounds; a cubic foot of water 62.4 pounds. If the stone is suspended in water it weighs only about 106.6 pounds, or 169 minus 62.4, its weight in air less the weight of one cubic foot of water.

If a body floats in a liquid it sinks until the displaced liquid weighs as much as the floating body.

Specific Gravity.—The specific gravity (sp. gr.) of a substance is the ratio of the weight of a certain volume of that substance to the weight of an equal volume of another substance adopted as a standard. In the case of solids and liquids the standard is pure water at a certain temperature, the various temperatures given by different authors being 32° F., 39.1° F., 60° F. and 62° F. In the case of gases the standard is air at the same temperature and pressure.

Example.—A cubic foot of pure water at 39.1° F. weighs approximately 62.4 pounds. A cubic foot of steel weighs 487.5 pounds. What is the specific gravity of the steel?

Solution.—Specific gravity $= \frac{487.5}{62.4} = 7.8.$

The table on specific gravity, weight, and specific heat of various solids given on pages 42 to 44 is a compilation of figures from several sources. The values vary, sometimes considerably, with different authors, and those given are a fair average. The values of specific heat are mostly for temperatures from 32 to 70° F.

Density.—By density of a substance is meant the weight per unit of volume. The density can thus be expressed in various units, such as pounds per cubic foot, pounds per cubic inch, etc.

Hydrometers.—Hydrometers are instruments used for determining the specific gravity or density of liquids. They are closed vessels filled with air and made so as to float vertically in a liquid, by loading their lower end with mercury or lead. The less the density of the liquid, the deeper the instrument sinks, and by having the stem graduated accordingly, the density of a liquid can be found.

Hydrometers are most commonly used for determining the concentration of different solutions. According to the purpose for which they are used, hydrometers are known as *salinometers,* for brine, *saccharometers,* for wort or other sugar solutions, *alcoholometers,* etc. The grading either indicates directly the percentage of salt, sugar, etc., in the solution, or the specific gravity, or is an entirely arbitrary scale, the readings of which have to be explained by tables. The ordinary or Balling saccharometer shows the per cent of extract in the wort. The salinometer or salometer indicates the per cent of salt in a brine solution.

Specific Gravity, Weight, and Specific Heat of Various Solids.

Substance.	Specific Gravity.	Weight per Cubic Foot in Pounds.	Specific Heat.
Aluminium	2.58—2.69	161—168	0.21
Antimony	6.70—6.86	418—428	0.0507
Asbestos	2.3	143.5	.....
Ashes	0.55	34.3	0.20
Asphaltum	1.39	86.7	.....
Bismuth	9.70—9.83	605.3—613.4	0.0308
Bone	1.8 —2.0	112—124	.....
Brass	7.8 —8.6	486.7—536.6	0.0939
Brick, common	1.8	112.3	
Brick, fire	2.24—2.4	140—150	0.20
Brick, pressed	2.16	135	
Bronze	8.52—8.96	532—559	.....
Butter	0.94	58.7	.....
Cadmium	8.56—8.70	534.1—542.9	.....
Cement, Portland	3.05—3.15	190.3—196.6	.....
Cement, Portland, loose.	1.25—1.51	78—94.2	.....
Chalk, air-dried	2.46—2.55	155.5—159.1	0.2148
Charcoal	.....	15—30	0.2415
Clay	1.92—2.4	120—150	.....
Coal, anthracite	1.60	100	0.201
Coal, bituminous	1.24—1.44	77—90	0.20
Cobalt	8.80	549.1	.....
Coke	0.7	44	0.2031
Coke, loose	.....	23—32	.....
Concrete	1.92—2.48	120—155	.....
Copper	8.81—8.94	549.7—557.9	0.0951
Cork	0.25	15.6	.....
Cotton	1.95	121.7	.....
Cypress (average)	0.53	33	.....
Earth, loose	1.15—1.28	72—80	.....
Earth, loamy	1.6	100	.....
Emery	4.0	250	.....
Fat, beef	0.925	57.7	.....
Glass, crown	2.5	156	0.161
Glass, plate	2.7	168.5	0.20
Glass, white flint	2.88—3.14	180—196.	0.117
Gold	19.37—19.55	1209—1220	0.0324
Granite	2.56—2.72	160—170	0.20
Graphite	2.20	137.3	0.202
Gravel	1.6—1.92	100—120	0.20

Substance.	Specific Gravity.	Weight per Cubic Foot in Pounds.	Specific Heat.
Gun metal	8.78	549	.....
Gutta percha	.97	60.5	.....
Gypsum	2.08—2.4	130—150	0.259
Ice at 32° F.	0.92	57.4	0.05040
India rubber	.93	58	.....
Iridium	22.38	1397	0.1887
Iron, pure	7.5—7.84	468—489.2	.....
Iron, cast (average)	7.2	450	0.1298
Iron, wrought (average)	7.7	480	0.1138
Ivory	1.83	114	.....
Lard	0.95	59.3	.....
Lead	11.35—11.44	708—714	0.314
Lime, quick (in bulk)	0.8—0.96	50—60	0.2169
Limestone	2.30—2.90	143.5—181	0.215
Magnesium	1.69—1.75	105.5—109.2	0.25
Manganese	7.4—8.0	462—499	0.1441
Maple, dry (average)	0.68	42	.....
Marble	2.56—2.88	160—180	0.2129
Masonry, dressed	2.24—2.88	140—180	0.20
Mercury	13.59—13.6	848—848.6	0.033
Mica	2.80	175	.....
Mortar	1.44—1.6	90—100	.....
Mud, dry	1.28—1.7	80—110	.....
Mud, fluid	1.67—1.92	104—120	.....
Nickel	8.27—8.93	516—557	0.1086
Oak, white (average)	0.77	48	0.57
Peat, condensed	1.0—1.3	62.4—81.1	.....
Pewter	7.47	466	.....
Phosphorus	1.836	114.6	0.25
Pine, white (average)	0.45	28	.....
Pine, yellow (average)	0.61	38	0.65
Pitch	1.15	72	.....
Plaster of Paris	1.50—1.81	93—113	0.1966
Platinum	20.85—21.71	1301—1355	0.0324
Porcelain, Chinese	2.385	148.8	.....
Potassium	0.865	54	.....
Quartz	2.64	165	0.188
Salt, rock	2.1—2.26	131—141	.....
Salt, table	2.13	133	.....
Sand	1.44—1.76	90—110	.....
Silver	10.51—10.61	656—662	0.0570
Slate	2.72—2.88	170—180	.....
Soapstone	2.65—2.8	166—175	.....
Sodium	0.973	60.7	.....

Substance.	Specific Gravity.	Weight per Cubic Foot in Pounds.	Specific Heat.
Steel	7.84	490	0.1165
Stone, various	2.16—3.4	135—200	.....
Sulphur	2.06	128.6	0.2026
Tar	1.0	62.4	.....
Tallow	0.94	58.6	.....
Tile	1.76—1.92	110—120	.....
Tin	7.29	455	0.0562
Tungsten	19.13	1194	.....
Wax, bee's	0.97	60.5	.....
White lead	3.16	197	.....
Wool	1.61	100.4	.....
Zinc	6.91—7.19	431.2—448.7	0.0956

Specific Gravity, Boiling Point at Atmospheric Pressure, and Specific Heat of Various Liquids.

Substance.	Specific Gravity.	Boiling Point, Degrees F.	Specific Heat.
Acetic Acid	1.057	246	0.51
Alcohol, grain	0.79	173	0.58
Alcohol, wood	0.81	150	0.60
Ammonia, liquid	0.623	—27	1.1 (av.)
Aqua Ammonia (28%)	0.891	...	...
Beer (average)	1.02	...	...
Beer Wort (13% extract)	1.0530	212	0.909
Benzine	0.7	195—212	0.45
Cider	1.02	...	...
Ethyl Ether	0.74	100	0.54
Glycerin	1.26	554	0.58
Linseed Oil	0.94	597	...
Milk	1.03	...	...
Mineral Oil, heavy	0.90—0.93	...	...
Mineral Oil, light	0.7	...	...
Olive Oil	0.92	412	0.40
Petroleum	0.79—0.82	316	0.50
Resin Oil	0.97	...	...
Salt Solution, saturated	1.27	226	...
Sea-Water (normal)	1.02—1.03	213.2	0.94
Sulphuric Acid	1.84	615	0.3350
Turpentine	0.87	315	0.42
Vinegar	1.08	...	0.92
Water	1.00	212	1.00

Specific Gravity and Balling Saccharometer Compared.

Per Cent Balling.	Specific Gravity.	Per Cent Balling.	Specific Gravity.	Per Cent Balling.	Specific Gravity.
0.5	1.0020	10.2	1.0412	12.9	1.0526
1.0	1.0040	10.4	1.0421	13.0	1.0530
1.5	1.0060	10.6	1.0429	13.1	1.0534
2.0	1.0080	10.8	1.0438	13.2	1.0538
2.5	1.0100	11.0	1.0446	13.3	1.0543
3.0	1.0120	11.1	1.0450	13.4	1.0547
3.5	1.0140	11.2	1.0454	13.5	1.0551
4.0	1.0160	11.3	1.0459	13.6	1.0555
4.5	1.0180	11.4	1.0463	13.7	1.0559
5.0	1.0200	11.5	1.0467	13.8	1.0564
5.5	1.0220	11.6	1.0471	13.9	1.0568
6.0	1.0240	11.7	1.0475	14.0	1.0572
6.5	1.0260	11.8	1.0480	14.5	1.0593
7.0	1.0281	11.9	1.0484	15.0	1.0614
7.5	1.0301	12.0	1.0488	15.5	1.0636
8.0	1.0322	12.1	1.0492	16.0	1.0657
8.5	1.0342	12.2	1.0496	16.5	1.0679
9.0	1.0363	12.3	1.0501	17.0	1.0700
9.2	1.0371	12.4	1.0505	17.5	1.0722
9.4	1.0379	12.5	1.0509	18.0	1.0744
9.6	1.0388	12.6	1.0513	18.5	1.0766
9.8	1.0396	12.7	1.0517	19.0	1.0788
10.0	1.0404	12.8	1.0522	19.5	1.0810

Comparative Table of Beaumé Degrees and Specific Gravity According to Bourgougnon.

For Liquids Heavier Than Water.

Deg. B.	Sp. Gr.	Deg. B.	Sp. Gr.	Deg. B.	Sp. Gr.	Deg. B.	Sp. Gr.
0	1.0000	19	1.1516	38	1.3574	57	1.6527
1	1.0069	20	1.1608	39	1.3703	58	1.6719
2	1.0140	21	1.1702	40	1.3834	59	1.6915
3	1.0212	22	1.1798	41	1.3968	60	1.7115
4	1.0285	23	1.1895	42	1.4104	61	1.7321
5	1.0358	24	1.1994	43	1.4244	62	1.7531
6	1.0433	25	1.2095	44	1.4386	63	1.7748
7	1.0509	26	1.2197	45	1.4530	64	1.7968
8	1.0586	27	1.2301	46	1.4678	65	1.8194
9	1.0665	28	1.2407	47	1.4829	66	1.8427
10	1.0744	29	1.2514	48	1.4983	67	1.8665
11	1.0825	30	1.2624	49	1.5140	68	1.8909
12	1.0906	31	1.2735	50	1.5301	69	1.9161
13	1.0989	32	1.2849	51	1.5465	70	1.9418
14	1.1074	33	1.2964	52	1.5632	71	1.9683
15	1.1159	34	1.3081	53	1.5802	72	1.9955
16	1.1246	35	1.3201	54	1.5978	73	2.0235
17	1.1335	36	1.3323	55	1.6157	74	2.0523
18	1.1424	37	1.3447	56	1.6340	75	2.0819

For Liquids Lighter Than Water.

Deg. B.	Sp. Gr.	Deg. B.	Sp. Gr.	Deg. B.	Sp. Gr.	Deg. B.	Sp. Gr.
10	1.000	23	0.918	36	0.849	49	0.789
11	0.993	24	0.913	37	0.844	50	0.785
12	0.986	25	0.907	38	0.839	51	0.781
13	0.980	26	0.901	39	0.834	52	0.777
14	0.973	27	0.896	40	0.830	53	0.773
15	0.967	28	0.890	41	0.825	54	0.768
16	0.960	29	0.885	42	0.820	55	0.764
17	0.954	30	0.880	43	0.816	56	0.760
18	0.948	31	0.874	44	0.811	57	0.757
19	0.942	32	0.869	45	0.807	58	0.753
20	0.936	33	0.864	46	0.802	59	0.749
21	0.930	34	0.859	47	0.798	60	0.745
22	0.924	35	0.854	48	0.794		

Efflux of Liquids.

When water flows out of an opening in the bottom of a vessel, it flows under a *head* equal to the total height of water in the vessel. When water flows out of an opening in the side of the vessel, the head is the height from center of the opening to the level of the water. In both cases the theoretical velocity of the outflowing water is:

$$v = \sqrt{2g\,h.}$$

v = velocity of outflow in feet per minute.
g = 32.16.
h = head in feet.

Since $\sqrt{2g} = \sqrt{2 \times 32.16} = 8.02$, the formula may be expressed as $v = 8.02\ \sqrt{h.}$

It will be noticed that the velocity is the same as that which would be acquired by a body falling freely through a height equal to the head of water. Also in this case the velocity is independent of the specific gravity of the liquid.

The *theoretical discharge* in cubic feet per second is equal to the product of the theoretical velocity in feet per second, and the area of the opening in square feet. The actual discharge is always 20 to 40 per cent less, as shown later.

The following formula expresses the theoretical discharge:

$$Q = a\ \sqrt{2g\,h.}$$

Q = theoretical discharge in cubic feet per second.
a = area of orifice in square feet.
g = 32.16.
h = head in feet.

In flowing out through an orifice the jet of water is contracted, forming the so-called contracted vein. Hence, in calculating the *actual velocity of discharge,* the theoretical result as obtained above must be multiplied by the *coefficient of discharge,* representing this contraction, which has in many cases been approximately determined by experiment. In emptying a tank, the head decreases as the level of the liquid falls. The form of the orifice also should be considered, a *standard orifice,* whose edges are straight, causing a greater contraction of the jet and reducing the theoretical outflow. For this reason rounded edges are often used.

The *coefficient of discharge* through a thin wall (no rounded corners) is known to be $m = 0.62$. The quantity of discharge per second in cubic feet is therefore:

$$Q = 0.62 \times a \times \sqrt{2g\,h.}$$

In case of very well rounded and tapered outflow pieces, taking the discharge area at the beginning of the cone, the contraction coefficient is $m = 0.82$.

Example.—How many gallons of water flow per minute through an orifice of 4 inches diameter, with sharp edges, under a head of 100 feet?

Solution.—$$Q = \frac{0.62 \times 4^2 \times 0.7854 \times \sqrt{2 \times 32.16 \times 100}}{144}$$

$= 4.339$ cubic feet per second.

The *hydraulic pressure* in pounds per square inch can be found from the head in feet, and vice versa, by means of the following formulas:

Pressure per square inch $= 0.433 \times$ head in feet.

Head in feet $= 2.309 \times$ pressure per square inch.

Velocity in feet per second $= 12.2 \sqrt{\text{pressure per sq. in.}}$

Time to Empty a Tank.—The following formula for calculating the time for emptying a tank is derived from that for quantity of discharge through a rounded orifice, using a coefficient of discharge of 0.82.

$$T = 0.30 \times \frac{A}{a} \times \sqrt{H.}$$

T = time in seconds for emptying the tank through rounded orifice.

A = area of base of tank in square feet.

a = area of the rounded orifice near the bottom in square feet.

H = depth of water in tank in feet.

Example.—The water in an upright cylindrical tank is 5 feet deep, and the diameter of the tank is 8 feet. How long will it take to run off the liquid through a standard tap hole 1.5 inches in diameter, opening into a straight, horizontal pipe of suitable width?

Solution.—

$A = 8^2 \times 0.7854 = 50.27$ square feet.

$$a = \frac{1.5^2 \times 0.7854}{144} = 0.01227 \text{ square foot.}$$

$H = 5$ feet.

$$T = 0.30 \times \frac{50.27}{0.01227} \times \sqrt{5} = 2{,}748 \text{ seconds.}$$

The actual time required is 2,748 seconds, or 45 minutes, 48 seconds.

Flow through Pipes.—The quantity of discharge of water through a pipe depends on various factors, i. e., the head, the length and diameter of pipe, smoothness of inside, and the number and kind of bends. In general, it may be said about the *loss due to friction,* that:

1. It is proportional to the length of the pipe.
2. It increases nearly as the square of the velocity.
3. It decreases as the diameter of the pipe increases.
4. It increases with the roughness of the interior surface.
5. It is independent of the pressure of the water.

Though the formulas for calculating the loss of head due to friction vary, the following gives approximate results in the case of a straight pipe:

$$h = 0.024 \times \frac{1 \times v^2}{d \times 2g}$$

$h =$ loss in head in feet.
$1 =$ length of pipe in feet.
$v =$ velocity of liquid in feet per second.
$d =$ diameter of pipe in feet.
$g = 32.16$.

The loss of head due to friction in *tees, angles* and *sharp bends* in general may be expressed by the formula:

$$h = k \times \frac{v^2}{2g}$$

For 45° angle, $k = 0.182$.
For 90° angle, $k = 0.98$.
For globe valve, $k = 1.91$.
For angle valve, $k = 2.94$.

A loss of head is suffered at the *inlet* to the pipe in the tank, varying according to the conditions, as the pipe may be flush with the bottom or projecting up in the tank, and may

have rough edges or be cut off smoothly. This loss is usually given as corresponding to a friction loss in a pipe length of 26 to 40 diameters. Under the best conditions, therefore, this loss may be placed at:

$$h = 0.024 \times \frac{26 \times d}{d} \times \frac{v^2}{2g} = 0.62 \frac{v^2}{2g}$$

Example.—The two tanks shown in Figure 32 are each 12 feet in diameter and have the same water level. The tank on the left has 16 feet of water above the center of the outlet, which is near the bottom, whereas the tank on the right has 6 feet of water, the outflow taking place through a vertical pipe 10 feet long, then an elbow, and 2 feet of horizontal pipe. How long will it take in each case to lower the water level 6 feet?

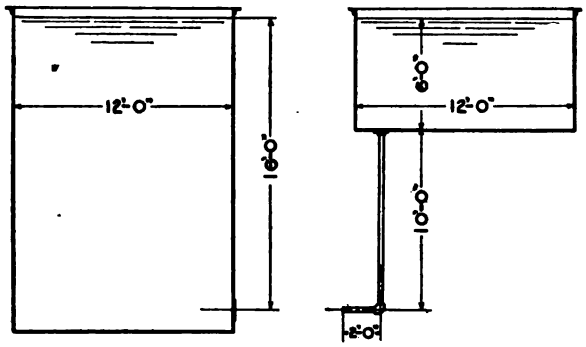


Figure 32. Flow of Liquids.

Solution.—Disregarding the outflow contraction and all frictional resistance, we would have for both cases (see page 46):

Maximum theoretical velocity, at the start:

$v = \sqrt{2 \times 32.16 \times 16} = 32.08$ feet per second.

Minimum theoretical velocity, at the moment the water level has been lowered 6 feet:

$v = \sqrt{2 \times 32.16 \times 10} = 25.36$ feet per second.

Average theoretical velocity:

$$\frac{32.08 + 25.36}{2} = 28.72 \text{ feet per second.}$$

The total amount of water flowing out is in both cases:

$12 \times 12 \times 0.7854 \times 6$ cubic feet.

The theoretical average flow per second is in both cases:

$$\frac{2 \times 2 \times 0.7854}{144} \times 28.72 \text{ cubic feet.}$$

The theoretical time for lowering the water 6 feet is then, in both cases:

$$12 \times 12 \times 0.7854 \times 6 \div \frac{2 \times 2 \times 0.7854}{144} \times 28.72 =$$

$$\frac{12 \times 12 \times 0.7854 \times 6 \times 144}{2 \times 2 \times 0.7854 \times 28.72} = 1{,}083 \text{ seconds} = 18 \text{ minutes, 3 seconds.}$$

In the case of the deep tank, with direct outflow, the theoretical figure will come true, except for the outflow contraction. Therefore, for this case the theoretical outflow should be multiplied by 0.62 (see page 47), giving as the actual average flow per second:

$$\frac{2 \times 2 \times 0.7854}{144} \times 28.72 \times 0.62 \text{ cubic feet.}$$

Inserting this in the above, gives as the actual time for lowering the water 6 feet, in the case of the deep tank:

$$\frac{1{,}083}{0.62} = 1{,}747 \text{ seconds} = 29 \text{ minutes, } 7 \text{ seconds.}$$

In the case of the tank with outflow through pipe, the actual outflow will differ much more from the theoretical.

The theoretical average velocity of 28.72 would require a head for overcoming friction in the straight pipe alone of (see formula page 48):

$$h = 0.024 \times \frac{12}{1/6} \times \frac{28.72^2}{2 \times 32.16} = 22 \text{ feet.}$$

To find the actual average velocity we must reason as follows:

At the start the head is 16 feet, and at the end of the outflow it is 10 feet, the average head being 13 feet. The unknown velocity obtained we call v, and we know that this head is used up in the following way (see page 48).

For overcoming pipe friction, $h_1 = 0.024 \times \frac{(10+2) \times v^2}{1/6 \times 2g}$

$= 1.73 \frac{v^2}{2g}$.

For loss at entrance to pipe, $h_2 = 0.62 \times \frac{v^2}{2g}$.

For loss in elbow, $h_3 = 0.98 \times \frac{v^2}{2g}$.

Left for producing velocity, $h = \frac{v^2}{2g}$.

$H = h_1 + h_2 + h_3 + h = 13.$

$h = 13 - h_1 - h_2 - h_3.$

$$\frac{v^2}{2g} = 13 - 1.73\,\frac{v^2}{2g} - 0.62\,\frac{v^2}{2g} - 0.98\,\frac{v^2}{2g}.$$

$$\frac{v^2}{2g} + 1.73\,\frac{v^2}{2g} + 0.62\,\frac{v^2}{2g} + 0.98\,\frac{v^2}{2g} = 13.$$

$$4.33 \times \frac{v^2}{2g} = 13.$$

$$\frac{v^2}{2g} = 13 \times 0.23.$$

$v^2 = 13 \times 0.23 \times 2 \times 32.16 = 193.11.$

$v = 13.90$ feet per second.

The actual time for flowing out of the 6 feet of water, from the tank provided with pipe, is:

$12 \times 12 \times 0.7854 \times 6 \div \frac{2 \times 2 \times 0.7854}{144} \times 13.90 = 2238$ seconds $= 37$ minutes, 18 seconds.

It is thus seen that, in this case, less than one-fourth of the head is used for producing velocity, the balance being absorbed by the work of overcoming the various resistances.

MECHANICS OF GASES.

Pascal's Law applies to gases as well as to liquids. In other words, the pressure of a gas on a surface is at right angles to the surface, and pressure applied at any point is equally transmitted in all directions. Archimedes' principle also applies to gases. In fact, gases have similar properties as liquids in every respect.

Atmospheric Pressure.

The pressure exerted by the atmosphere is an important example of gas pressure.

Air, being a material substance, has weight. The surface of the earth is the bottom of an ocean of air, and supports the weight of the entire atmosphere. Every square inch of the earth's sur-

face is, therefore, exposed to a certain pressure, which is equal to the weight of the column of air resting on it, and reaching from the surface of the earth up to where the air ends. The lower strata of the air, i. e., those nearest to the ground, are much denser and heavier than the upper ones, as they are compressed by the weight of the upper strata. The atmospheric pressure is, therefore, less on the top of a high mountain than it is at sea level, since the column of air over the top of the mountain is both shorter and less dense.

At sea level the pressure of the atmosphere is 14.7 pounds per square inch. This pressure is exerted on all bodies with equal power in all directions. In a bottle filled with air the pressure is equal on the outside and on the inside, but if the air be withdrawn from the bottle the outside pressure is no longer balanced by any pressure on the inside, and the walls of the bottle may break.

When the air is partly or entirely removed from a vessel there is said to be a partial or complete *vacuum* in the vessel.

If one end of an open tube is dipped into water, and the air sucked out at the other end, the water will immediately rise into the tube, and if the air is drawn out completely the water will rise to a height of nearly 34 feet, and no more, because the weight of a column of water of such height will balance the weight of the air. From this we can calculate that 1 pound of pressure corresponds to $\frac{34}{14.7} = 2.3$ feet of water. More exactly, one pound of pressure is equivalent to a head of water of 2.3093 feet or 27.71 inches, and one atmosphere (14.7 pounds per square inch) is equivalent to a head of water of 33.947 feet or 10.347 meters. The table below gives the pressures corresponding to various heads of water:

Pressure in Pounds Corresponding to Various Heads of Water.

Head in Feet.	Pressure in Pounds Per Square In.	Head in Feet.	Pressure in Pounds Per Square In.	Head in Feet.	Pressure in Pounds Per Square In.	Head in Feet.	Pressure in Pounds per Square inch
1	0.43	20	8.66	55	23.83	90	38.90
2	0.88	25	10.83	60	25.90	95	41.07
3	1.30	30	12.	65	28.06	100	43.33
4	1.74	35	15.16	70	30.55	125	54.17
5	2.16	40	17.33	75	32.72	150	65.
10	4.33	45	19.50	80	34.66	175	76.05
15	6 50	50	21.66	85	36.83	200	86.67

The following table gives the weight of water in pounds per cubic foot at various temperatures, and shows the maximum density at 39.1° F.:

Weight of Water in Pounds Per Cubic Foot at Various Temperatures.*

Temp. F.	Weight.	Temp. F.	Weight.	Temp. F.	Weight.
32	62.418	212	59.76	420	52.6
39.1	62.425	220	59.63	430	52.2
40	62.42	230	59.37	440	51.7
50	62.42	240	59.11	450	51.2
60	62.37	250	58.83	460	50.7
62	62.35	260	58.55	470	50.2
70	62.30	270	58.26	480	49.7
80	62.22	280	57.96	490	49.2
90	62.11	290	57.65	500	48.7
100	62.00	300	57.33	510	48.1
110	61.86	310	57.00	520	47.6
120	61.71	320	56.66	530	47.0
130	61.55	330	56.30	540	46.3
140	61.38	340	55.94	550	45.6
150	61.18	350	55.57	560	44.9
160	61.00	360	55.18	570	44.1
170	60.80	370	54.78	580	43.3
180	60.50	380	54.36	590	42.6
190	60.36	390	53.94	600	41.8
200	60.12	400	53.5		
210	59.88	410	53.0		

Siphon.—A siphon is a bent tube used to remove liquids from a vessel by means of atmospheric pressure.

Figure 33 represents a siphon having the two legs B A and B C. If the siphon is filled with water, it will be readily seen that there will be more water in B C than in B A, consequently the water in the latter will be heavier and will descend, thereby tending to create a vacuum or suction in the tube. This vacuum is immediately relieved, however, by the atmospheric pressure (represented by the arrows) forcing the water upwards through the opening H into the siphon. This flow will continue until the liquid stands at equal heights in both vessels, as shown by the

* Landolt & Börnstein's Tables.

dotted line D, or until the liquid is separated by air getting into the siphon at any point along the tube above the line A K.

Merely placing a siphon into a liquid will not cause the latter to flow; it is necessary first to prime the siphon, which is done either by filling it completely with liquid or by applying suction at the opening of the long end at C until the liquid begins to flow.

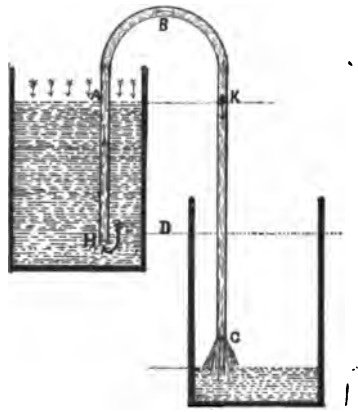


Figure 33. Syphon.

Barometer.—A barometer is an instrument used to measure atmospheric pressure.

The *mercury-barometer* is a glass tube about 36 inches high, the upper end of which is exhausted and closed air-tight, while the lower end is immersed in a vessel of mercury. The atmospheric pressure keeps the tube filled with mercury to a height of about 30 inches, the remaining 6 inches at the top being a complete vacuum. The atmospheric pressure, like the temperature of the air, is constantly changing, and these changes are shown by the rise or fall of the column of mercury in the glass tube.

The *aneroid-barometer* is a portable instrument, without any liquid. It consists of a metallic box, exhausted of air, the walls of which yield under the varying pressure of the air, their movements being transferred to an indicator on a graduated scale.

Law of Boyle-Mariotte.

The volume of a gas depends upon the pressure exerted upon it, the temperature remaining constant. If the pressure is increased two, three or four times, the volume decreases at the same

rate, that is, the gas that under a certain pressure occupies one cubic foot will, when the pressure increases four times, occupy one-fourth of one cubic foot.

The general law for the relation between pressure, volume and temperature of a gas may be written in this form:

$$p \times v = R \times T.$$

p = absolute pressure of gas in pounds per square foot.

v = volume of one pound of gas in cubic feet.

R = gas constant (for air = 53.35).

T = absolute temperature in degrees F.

If the temperature remains constant, i. e., in an isothermal expansion or compression, the product $p \times v$ is also constant, or expressed in an equation,

$$pv = p_1v_1$$

$$\frac{p}{p_1} = \frac{v_1}{v}$$

p = absolute pressure of gas in pounds per square inch before expansion or compression.

p_1 = absolute pressure of gas after expansion or compression.

v = volume at pressure p.

v_1 = volume at pressure p_1.

The formulas given above represent the law of Boyle-Mariotte. The temperature remaining constant, the volumes of the same weight of gas at different pressures vary inversely as the pressures.

From these formulas may be obtained by transposition

$$p = \frac{p_1v_1}{v}$$

$$v = \frac{p_1v_1}{p}$$

Example.—Ninety cubic feet of air are compressed from atmospheric pressure to a gauge pressure of 75 pounds. What will be the volume?

Solution.—$v = \frac{p_1v_1}{p}$

$$v = \frac{14.7 \times 90}{75 + 14.7} = 14.7 \text{ cubic feet.}$$

Example.—One hundred and twenty cubic feet of air of 100

pounds gauge pressure are required. How many cubic feet of air at atmospheric pressure must be compressed.

Solution.—$v = \frac{p_1 v_1}{p}$

$$v = \frac{114.7 \times 120}{14.7} = 936 \text{ cubic feet.}$$

Both of these examples of course assume that the temperature remains constant. However, as gases on being compressed become heated, the actual volume will be higher, unless the gas is cooled to the original temperature.

Law of Gay-Lussac or Charles.

This law states: The volume of a gas changes directly proportional with its absolute temperature, when the pressure remains the same. The pressure of a gas changes directly proportional with its absolute temperature when the volume is kept the same.

Expressed in formulas, we have the following:

$$\frac{v}{v_1} = \frac{T}{T_1}$$

$$\frac{p}{p_1} = \frac{T}{T_1}$$

The signs v_1, p_1 and T_1 are the respective values of v, p and T for another condition.

The above laws combined give the equation:

$$\frac{pv}{T} = \frac{p_1 v_1}{T_1}$$

These laws hold only for a perfect gas, but though there are deviations, they are sufficiently accurate for all practical purposes.

In the equation $pv = RT$, the gas constant for any permanent gas may be determined from the volume of a pound of the gas at a certain temperature and pressure. For instance, for air, it is found as follows:

At atmospheric pressure and 32° F., one pound of air has a volume of 12.387 cubic feet. Expressed in absolute temperature, 32° F. is 492 degrees, and the atmospheric pressure is $14.7 \times 144 = 2{,}116.5$ pounds per square foot. We therefore have that the constant R for air is:

$$R = \frac{12.387 \times 2{,}116.5}{492} = 53.3.$$

HEAT.

The molecules of all bodies are in continual motion, and this motion is what we call heat. The more rapidly the molecules move, the hotter is the body. If the movement of the molecules should stop entirely the body would have no heat; it would be absolutely cold. Calculations for the gases have led to the conclusion that such a state would be reached at about 460 degrees below the zero point of a Fahrenheit thermometer, this point being called the *absolute zero.*

When two bodies of different temperatures are brought in contact with each other, the temperature of the warmer one falls while that of the colder one rises, that is, the rapidly moving particles of the warmer body cause the particles of the colder body to move more rapidly than before, while the movements of its own particles are lessened. This process takes place in a similar way as when a fast moving train runs from the rear into one of less speed; the slow train is pushed ahead faster, while the speed of the other is lessened.

The greatest *source of heat* is the sun, but heat can also be produced by chemical changes, as illustrated by the burning of coal, or by friction.

Measurement of Heat.

Thermometers.—Thermometers measure the degree of heat. The warmer a body is, the larger is its volume, and an increase in volume indicates a corresponding rise in temperature; a contraction of volume indicates a corresponding fall in temperature. This property is utilized for measuring temperatures. The substance most commonly used for thermometers is mercury.

For high temperatures, so-called *pyrometers* are used, of which there are a large number of varieties. So-called *recording thermometers* are described on page 209.

The fixed points of all thermometer scales are the temperatures of melting ice and of boiling water, both under standard atmospheric pressure (30 inches mercury). After these two points are marked off, the distance between them is divided up into 180 equal parts for the Fahrenheit scale, 100 equal parts for the Centigrade (Celsius) scale, and 80 equal parts for the Reaumur scale. The Centigrade and the Reaumur thermometers have the

Conversion Tables of the Thermometer Scales.

Fahrenheit to Reaumur and Celsius.

F.	R.	C.	F.	R.	C.	F.	R.	C.	F.	R.	C.
°	°	°	°	°	°	°	°	°	°	°	°
212	80.0	100	148	51.6	64.4	84	23.1	28.9	20	— 5.3	— 6.7
211	79.6	99.4	147	51.1	63.9	83	22.7	28.3	19	— 5.8	— 7.2
210	79.1	98.9	146	50.7	63.3	82	22.2	27.8	18	— 6.2	— 7.8
209	78.7	98.3	145	50.2	62.8	81	21.8	27.2	17	— 6.7	— 8.3
208	78.2	97.8	144	49.8	62.2	80	21.3	26.7	16	— 7.1	— 8.9
207	77.8	97.2	143	49.3	61.7	79	20.9	26.1	15	— 7.6	— 9.4
206	77.3	96.7	142	48.9	61.1	78	20.4	25.6	14	— 8.0	—10.0
205	76.9	96.1	141	48.4	60.6	77	20.0	25.0	13	— 8.4	—10.6
204	76.4	95.6	140	48.0	60.0	76	19.6	24.4	12	— 8.9	—11.1
203	76.0	95.0	139	47.6	59.4	75	19.1	23.9	11	— 9.3	—11.7
202	75.6	94.4	138	47.1	58.9	74	18.7	23.3	10	— 9.8	—12.2
201	75.1	93.9	137	46.7	58.3	73	18.2	22.8	9	—10.2	—12.8
200	74.7	93.3	136	46.2	57.8	72	17.8	22.2	8	—10.7	—13.3
199	74.2	92.8	135	45.8	57.2	71	17.3	21.7	7	—11.1	—13.9
198	73.8	92.2	134	45.3	56.7	70	16.9	21.1	6	—11.6	—14.4
197	73.3	91.7	133	44.9	56.1	69	16.4	20.6	5	—12.0	—15.0
196	72.9	91.1	132	44.4	55.6	68	16.0	20.0	4	—12.4	—15.6
195	72.4	90.6	131	44.0	55.0	67	15.6	19.4	3	—12.9	—16.1
194	72.0	90.0	130	43.6	54.4	66	15.1	18.9	2	—13.3	—16.7
193	71.6	89.4	129	43.1	53.9	65	14.7	18.3	1	—13.8	—17.2
192	71.1	88.9	128	42.7	53.3	64	14.2	17.8	0	—14.2	—17.8
191	70.7	88.3	127	42.2	52.8	63	13.8	17.2	—1	—14.7	—18.3
190	70.2	87.8	126	41.8	52.2	62	13.3	16.7	—2	—15.1	—18.9
189	69.8	87.2	125	41.3	51.7	61	12.9	16.1	—3	—15.6	—19.4
188	69.3	86.7	124	40.9	51.1	60	12.4	15.6	—4	—16.0	—20.0
187	68.9	86.1	123	40.4	50.6	59	12.0	15.0	—5	—16.4	—20.6
186	68.4	85.6	122	40.0	50.0	58	11.6	14.4	—6	—16.9	—21.1
185	68.0	85.0	121	39.6	49.4	57	11.1	13.9	—7	—17.3	—21.7
184	67.6	84.4	120	39.1	48.9	56	10.7	13.3	—8	—17.8	—22.2
183	67.1	83.9	119	38.7	48.3	55	10.2	12.8	—9	—18.2	—22.8
182	66.7	83.3	118	38.2	47.8	54	9.8	12.2	—10	—18.7	—23.3
181	66.2	82.8	117	37.8	47.2	53	9.3	11.7	—11	—19.1	—23.9
180	65.8	82.2	116	37.3	46.7	52	8.9	11.1	—12	—19.6	—24.4
179	65.3	81.7	115	36.9	46.1	51	8.4	10.6	—13	—20.0	—25.0
178	64.9	81.1	114	36.4	45.6	50	8.0	10.0	—14	—20.4	—25.6
177	64.4	80.6	113	36.0	45.0	49	7.6	9.4	—15	—20.9	—26.1
176	64.0	80.0	112	35.6	44.4	48	7.1	8.9	—16	—21.3	—26.7
175	63.6	79.4	111	35.1	43.9	47	6.7	8.3	—17	—21.8	—27.2
174	63.1	78.9	110	34.7	43.3	46	6.2	7.8	—18	—22.2	—27.8
173	62.7	78.3	109	34.2	42.8	45	5.8	7.2	—19	—22.7	—28.3
172	62.2	77.8	108	33.8	42.2	44	5.3	6.7	—20	—23.1	—28.9
171	61.8	77.2	107	33.3	41.7	43	4.9	6.1	—21	—23.6	—29.4
170	61.3	76.7	106	32.9	41.1	42	4.4	5.6	—22	—24.0	—30.0
169	60.9	76.1	105	32.4	40.6	41	4.0	5.0	—23	—24.4	—30.6
168	60.4	75.6	104	32.0	40.0	40	3.6	4.4	—24	—24.9	—31.1
167	60.0	75.0	103	31.6	39.4	39	3.1	3.9	—25	—25.3	—31.7
166	59.6	74.4	102	31.1	38.9	38	2.7	3.3	—26	—25.8	—32.2
165	59.1	73.9	101	30.7	38.3	37	2.2	2.8	—27	—26.2	—32.8
164	58.7	73.3	100	30.2	37.8	36	1.8	2.2	—28	—26.7	—33.3
163	58.2	72.8	99	29.8	37.2	35	1.3	1.7	—29	—27.1	—33.9
162	57.8	72.2	98	29.3	36.7	34	0.9	1.1	—30	—27.6	—34.4
161	57.3	71.7	97	28.9	36.1	33	0.4	0.6	—31	—28.0	—35.0
160	56.9	71.1	96	28.4	35.6	32	0.0	0.0	—32	—28.4	—35.6
159	56.4	70.6	95	28.0	35.0	31	—0.4	—0.6	—33	—28.9	—36.1
158	56.0	70.0	94	27.6	34.4	30	—0.9	—1.1	—34	—29.3	—36.7
157	55.6	69.4	93	27.1	33.9	29	—1.3	—1.7	—35	—29.8	—37.2
156	55.1	68.9	92	26.7	33.3	28	—1.8	—2.2	—36	—30.2	—37.8
155	54.7	68.3	91	26.2	32.8	27	—2.2	—2.8	—37	—30.7	—38.3
154	54.2	67.8	90	25.8	32.2	26	—2.7	—3.3	—38	—31.1	—38.9
153	53.8	67.2	89	25.3	31.7	25	—3.1	—3.9	—39	—31.6	—39.4
152	53.3	66.7	88	24.9	31.1	24	—3.6	—4.4	—40	—32.0	—40.0
151	52.9	66.1	87	24.4	30.6	23	—4.0	—5.0			
150	52.4	65.6	86	24.0	30.0	22	—4.4	—5.6			
149	52.0	65.0	85	23.6	29.4	21	—4.9	—6.1			

Conversion Tables of the Thermometer Scales.

Reaumur to Fahrenheit and Celsius.

R.	F.	C.	R.	F.	C.	R.	F.	C.	R.	F.	C.
°	°	°	°	°	°	°	°	°	°	°	°
80	212.00	100.00	49	142.25	61.25	18	72.50	22.50	—13	2.75	—16.25
79	209.75	98.75	48	140.00	60.00	17	70.25	21.25	—14	0.50	—17.50
78	207.50	97.50	47	137.75	58.75	16	68.00	20.00	—15	— 1.75	—18.75
77	205.25	96.25	46	135.50	57.50	15	65.75	18.75	—16	— 4.00	—20.00
76	203.00	95.00	45	133.25	56.25	14	63.50	17.50	—17	— 6.25	—21.25
75	200.75	93.75	44	131.00	55.00	13	61.25	16.25	—18	— 8.50	—22.50
74	198.50	92.50	43	128.75	53.75	12	59.00	15.00	—19	—10.75	—23.75
73	196.25	91.25	42	126.50	52.50	11	56.75	13 75	—20	—13.00	—25.00
72	194.00	90.00	41	124.25	51.25	10	54.50	12.50	—21	—15.25	—26.25
71	191.75	88.75	40	122.00	50.00	9	52 25	11.25	—22	—17.50	—27.50
70	189.50	87.50	39	119.75	48.75	8	50.00	10.00	—23	—19.75	—28.75
69	187.25	86.25	38	117.50	47.50	7	47.75	8.75	—24	—22.00	—30.00
68	185.00	85.00	37	115.25	46.25	6	45 50	7.50	—25	—24.25	—31.25
67	182.75	83.75	36	113.00	45.00	5	43.25	6 25	—26	—26.50	—32.50
66	180.50	82.50	35	110.75	43.75	4	41 00	5.00	—27	— 28.75	—33.75
65	178.25	81.25	34	108.50	42.50	3	38.75	3.75	—28	—31.00	—35.00
64	176.00	80.00	33	106.25	41.25	2	36.50	2.50	—29	—33.25	—36.25
63	173.75	78.75	32	104.00	40.00	1	34.25	1.25	—30	—35.50	—37.50
62	171.50	77.50	31	101.75	38.75	0	32.00	0.00	—31	—37.75	—38.75
61	169.25	76.25	30	99.50	37.50	— 1	29.75	— 1.25	—32	— 40.00	—40.00
60	167.00	75.00	29	97.25	36.25	— 2	27.50	— 2.50	—33	—42.25	—41.25
59	164.75	73.75	28	95.00	35.00	— 3	25.25	— 3.75	—34	—44.50	—42.50
58	162.50	72.50	27	92.75	33.75	— 4	23.00	— 5.00	—35	—46.75	—43.75
57	160.25	71.25	26	90.50	32.50	— 5	20.75	— 6.25	—36	—49.00	—45.00
56	158.00	70.00	25	88.25	31.25	— 6	18.50	— 7.50	—37	—51.25	—46.25
55	155.75	68.75	24	86.00	30.00	— 7	16.25	— 8.75	—38	—53.50	—47.50
54	153.50	67.50	23	83.75	28.75	— 8	14.00	—10.00	—39	—55.75	—48.75
53	151.25	66.25	22	81.50	27.50	— 9	11.75	—11.25	—40	—58.00	—50.00
52	149.00	65.00	21	79.25	26.25	—10	9.50	—12.50			
51	146.75	63.75	20	77.00	25.00	—11	7.25	—13.75			
50	144.50	62.50	19	74.75	23.75	—12	5.00	—15.00			

Celsius to Reaumur and Fahrenheit.

C.	R.	F.	C.	R.	F.	C.	R.	F.	C.	R.	F.
°	°	°	°	°	°	°	°	°	°	°	°
100	80.0	212.0	72	57.6	161.6	44	35.2	111.2	16	12.8	60.8
99	79.2	210.2	71	56.8	159.8	43	34.4	109.4	15	12.0	59.0
98	78.4	208.4	70	56.0	158.0	42	33.6	107.6	14	11.2	57.2
97	77.6	206.6	69	55.2	156.2	41	32.8	105.8	13	10.4	55.5
96	76.8	204.8	68	54.4	154.4	40	32.0	104.0	12	9.6	53.6
95	76.0	203.0	67	53.6	152.6	39	31.2	102.2	11	8.8	51.8
94	75.2	204.2	66	52.8	150.8	38	30.4	100.4	10	8.0	50 0
93	74.4	199.4	65	52.0	149.0	37	29.6	98.6	9	7.2	48.2
92	73.6	197.6	64	51.2	147.2	36	28.8	96.8	8	6.4	46.4
91	72.8	195.8	63	50.4	145.4	35	28.0	95.0	7	5.6	44.6
90	72.0	194.0	62	49.6	143.6	34	27.2	93.2	6	4.8	42.8
89	71.2	192.2	61	48.8	141.8	33	26.4	91.4	5	4.0	41.0
88	70.4	190.4	60	48.0	140.0	32	25.6	89.6	4	3.2	39.2
87	69.6	188.6	59	47.2	138.2	31	24.8	87.8	3	2.4	37.4
86	68.8	186.8	58	46.4	136.4	30	24.0	86 0	2	1.6	35.6
85	68.0	185.0	57	45.6	134.6	29	23.2	84.2	1	0.8	33.8
84	67.2	183.2	56	44.8	132.8	28	22.4	82.4	0	0.0	32.0
83	66.4	181.4	55	44.0	131.0	27	21.6	80.6	—1	— 0.8	30.2
82	65.6	179.6	54	43.2	129.2	26	20.8	78.8	—2	—1.6	28.4
81	64.8	177.8	53	42.4	127.4	25	20.0	77.0	—3	—2.4	26.6
80	64.0	176.0	52	41.6	125.6	24	19.2	75.2	—4	—3.2	24.8
79	63.2	174.2	51	40.8	123.8	23	18.4	73.4	—5	—4.0	23.0
78	62.4	172.4	50	40.0	122.0	22	17.6	71.6	—6	—4.8	21.2
77	61.6	170.6	49	39.2	120.2	21	16.8	69.8	—7	—5 6	19.4
76	60.8	168.8	48	38.4	118.4	20	16.0	68.0	—8	—6.4	17.6
75	60.0	167.0	17	37.6	116.6	19	15.2	66.2	—9	—7.2	15.8
74	59.2	165.2	46	36.8	114.8	18	14.4	64.4	—10	—8.0	14.0
73	58.4	163.4	45	36.0	113.0	17	13.6	62.6	—11	—8.8	12 2

point of melting ice, which is also the freezing point, marked 0, so the point of boiling water is marked 100 degrees and 80 degrees respectively. The Fahrenheit thermometer has the point of melting ice marked 32 degrees, and the point of boiling water 212 degrees. In most European and Latin-American countries the Centigrade thermometer is the standard, whereas in this country the Fahrenheit is most commonly used, though the Centigrade is used considerably in scientific work. The Reaumur thermometer has its chief use in breweries.

The conversion from one thermometric scale to another is shown in the following simple rules. The tables on pages 58 and 59 may also be used for this purpose.

$$F = C \times \frac{9}{5} + 32. \quad F = R \times \frac{9}{4} + 32.$$

$$C = (F - 32) \times \frac{5}{9}. \quad R = (F - 32) \times \frac{4}{9}.$$

Example.—Change 50° Centigrade and 60° Reaumur into Fahrenheit.

Solution.—$F = 50 \times \frac{9}{5} + 32 = 122.$

$$F = 60 \times \frac{9}{4} + 32 = 167.$$

50° C.= 122° F. 60° R.= 167° F.

Example.—Change 140° F. into Reaumur and Centigrade.

Solution.—$C = (140 - 32) \times \frac{5}{9} = 108 \times \frac{5}{9} = 60°$ C.

$$R = (140 - 32) \times \frac{4}{9} = 108 \times \frac{4}{9} = 48°\text{ R.}$$

Transfer or Distribution of Heat.

Heat moves from one point to another in three ways: conduction, circulation (or convection), and radiation.

Conduction of Heat.—If one end of a bar of iron is held in a fire the temperature will soon rise at the other end of the bar. The heat of the fire has traveled from layer to layer of the bar without any sensible motion of the iron particles. Heat transmitted in this way is called conducted heat.

A stick of wood may be held in a fire without burning the hand, whereas with a bar of metal under similar circumstances

this could not be done. The reason for this is that the wood does not conduct the heat as well as do the metals.

Metals are the best conductors of heat, and of the metals silver is the best, then follow copper, gold, tin and iron. The next best conductors are stones, dense woods and charcoal, then liquids in general, then cork, wool, asbestos, called poor conductors or insulators, and finally, gases, which conduct hardly any heat at all.

Circulation of Heat.—Although liquids and gases are poor conductors of heat, they may, nevertheless, be quickly heated, if the heat is applied to the bottoms of the vessels containing them. This heating, however, is not by conduction, but by circulation.

A circulation is set up in the liquid or gas when the portions in contact with the bottom of the vessel get heated, become lighter and rise to the surface, carrying the heat with them.

Radiation of Heat.—There is another and entirely different way by which heat is transmitted through certain media, and travels with the speed of light, being, in fact, of the same nature as light. Such is the transmission of heat from the sun to the earth, and from any hot body through the air. In this case the heat does not to any extent affect the air through which it travels; it passes between the molecules of the air. Heat that moves in such manner is said to be radiated.

If radiated heat strikes a body too solid to allow it to pass through, the heat is absorbed and raises the temperature of that body.

The light and heat from the sun pass through the atmosphere without warming it, but they heat the earth. From the warm earth, the lower strata of air are then heated by conduction and circulation.

Expansion by Heat.

As stated before, when a body is heated, the molecules increase in speed, as a result of which there is an increase in their distance from each other, that is, the body expands. By expansion, the same mass occupies more space than before, and hence displaces more of the surrounding matter, and its specific gravity decreases.

Expansion of Solids.—Nearly all solid bodies expand

Coefficient of Expansion of Various Solids at Ordinary Temperatures.

Substance	Coefficient for 1° Fahr.
Aluminium (Cast)	.00001234
Antimony (Crystallized)	.00000627
Brass (Cast)	.00000957
Brass (English Plate)	.00001052
Brass (Sheet)	.00001040
Brick (Best Stock)	.00000306
Brick in Cement Mortar (Headers)	.00000494
Brick in Cement Mortar (Stretchers)	.00000256
Bronze	.00000975
Cement (Portland, Neat)	.00000594
Cement (Portland, with Sand)	.00000656
Cement (Roman, Dry)	.00000797
Copper	.00000887
Glass (Blown)	.00000498
Glass (Flint)	.00000451
Glass (Hard)	.00000397
Glass (Thermometer)	.00000499
Glass (White, Free from Lead)	.00000492
Granite (Gray, Dry)	.00000438
Granite (Red, Dry)	.00000498
Gold (Pure)	.00000786
Invar	.0000002+
Iron (Cast)	.00000556
Iron (Soft)	.00000626
Iron (Swedish)	.00000636
Iron (Wrought)	.00000648
Lead	.00001571
Marble (Ordinary, Dry)	.00000363
Marble (Ordinary, Moist)	.00000663
Mercury (Cubic Expansion)	.00009984
Nickel	.00000695
Plaster (White)	.00000922
Platinum	.00000479
Silver (Pure)	.00001079
Slate	.00000577
Steel (Cast)	.00000636
Steel (Tempered)	.00000689
Stone (Sand, Dry)	.00000652
Tin	.00001163
Wood (Pine)	.00000276
Zinc	.00001407
Zinc 8, Tin 1	.00001496

when heated and contract when cooled. The amount of expansion which different solids undergo depends on the nature of the solid, and the temperature. When heated from 32° F. to 212° F., the linear expansion in rods of the following substances is:

Steel, tempered.....0.001240 = $\frac{1}{806}$

Copper .....0.001596 = $\frac{1}{627}$

For instance, a bar of steel 806 inches long at 32° F. will be one inch longer, or 807 inches, when heated to 212° F. The above figures vary somewhat with different authors.

Expansion of Liquids.—The expansion of liquids varies greatly with the nature of the substance. Between 32° F. and 212° F. water expands 0.045 $\left(\frac{1}{22}\right)$ of its original volume, alcohol 0.111 $\left(\frac{1}{9}\right)$. Hence 22 barrels of water at 32° F. will fill 23 barrels at 212° F.

Water forms an exception to the uniformity of progressive expansion with rising temperatures. When heated from 32° F. to 39.1° F., it contracts instead of expanding, and only after reaching the temperature of 39.1° F. does it begin to expand when heated. The temperature of 39.1° F. marks its greatest density, and from this point it expands, whether it is heated or cooled.

Expansion of Gases.—All gases expand nearly alike for an equal increase in temperature, and the proportional increase in volume is independent of the temperature and pressure of the gas. The amount of expansion for a rise in temperature from 32° F. to 212° F. is 0.366 of the original volume.

Heat Units.

Definition of Heat Unit.—A heat unit is the amount of heat that will raise a unit amount of water one degree in temperature.

British Thermal Unit (B. T. U.).—The old definition of a B. T. U. as the quantity of heat required to raise one pound of water from 39.1 to 40.1° F. is no longer used. A recent definition and the one followed in this book is that it is $\frac{1}{180}$ of the heat required to raise the temperature of one pound of water from 32° F. to 212° F.

Other Heat Units.—Instead of taking for a unit the heat that can raise one pound of water one degree Fahrenheit, a heat unit may be calculated upon a kilogram of water, a barrel of water or any quantity that may be convenient, and instead of Fahrenheit degrees, Reaumur or Celsius (Centigrade) may be used.

Specific Heat.

Equal weights of different substances require different amounts of heat to raise them a given temperature. Water requires more heat than almost any other body, solid or liquid, and is, therefore, used as a unit. The figure that expresses how much heat is required to raise the temperature of a certain weight of a body one degree as compared with that required to raise the temperature of an equal weight of water by one degree, is called the specific heat of the substance.

The specific heat of cast iron, for instance, is 0.1298, that is, it takes only 0.1298 of the heat which will raise a certain weight

Specific Heat of Water at Various Temperatures.*

Degrees F.	Specific Heat	Degrees F.	Specific Heat	Degrees F.	Specific Heat
20	1.0168	220	1.007	420	1.072
30	1.0098	230	1.009	430	1.077
40	1.0045	240	1.012	440	1.082
50	1.0012	250	1.015	450	1.086
60	0.9990	260	1.018	460	1.091
70	0.9977	270	1.021	470	1.096
80	0.9970	280	1.023	480	1.101
90	0.9967	290	1.026	490	1.106
100	0.9967	300	1.029	500	1.112
110	0.9970	310	1.032	510	1.117
120	0.9974	320	1.035	520	1.123
130	0.9974	330	1.038	530	1.128
140	0.9986	340	1.041	540	1.134
150	0.9994	350	1.045	550	1.140
160	1.0002	360	1.048	560	1.146
170	1.0010	370	1.052	570	1.152
180	1.0019	380	1.056	580	1.158
190	1.0029	390	1.060	590	1.165
200	1.0039	400	1.064	600	1.172
210	1.0050	410	1.068		

* From Marks & Davis' "Steam Tables."

of water one degree to raise the temperature of an equal weight of cast iron one degree. To heat 10 pounds of water from 40 to 100° F. requires $10 \times 60 = 600$ B. T. U. To heat 10 pounds of cast iron from 40 to 100° F. requires $10 \times 60 \times 0.1298 = 77.88$ B. T. U.

The specific heats of various solids and liquids are given in the tables on pages 42 to 44, whereas those of gases are given on page 138. On the previous page is given a table on the specific heat of water at various temperatures.

Change of State.

Many solid bodies when heated are liquefied or fused, i. e., they melt. To effect the change from solid to liquid a certain amount of heat is required. If one pound of water of 32° F. and one pound of water of 176° F. are mixed, the temperature of the mixture will be 104° F., or half way between 32° F. and 176° F. If, however, we mix one pound of ice of 32° F. and one pound of water of 176° F. we will have two pounds of water of 32° F. That is, all the heat given off by the warm water has been just sufficient to melt the ice without producing any rise in temperature. Heat which is absorbed by melting bodies is called *latent heat of fusion.* The latent heat of one pound of water melted from ice is 144 B. T. U.

In an exactly similar way heat is absorbed when a liquid is changed into a vapor. If we place a thermometer in water that is being heated under atmospheric pressure, the thermometer will rise until it reaches 212° F., at which point it becomes stationary. Any further heat added does not raise the temperature of the boiling water but is entirely consumed in transforming the water into steam.

The heat which is absorbed by the steam is called the *latent heat of vaporization.* If steam is condensed to water, this absorbed heat is set free once more. One pound of steam of 212° F., when condensed by being conducted into cold water, gives out heat enough to raise 5.391 pounds of water from 32° F. to 212° F. The latent heat of vaporization of one pound of steam at atmospheric pressure therefore is $5.391 \times 180 = 970.4$ B. T. U.

Distillation.—If a liquid is made to boil, and the vapors are condensed again, the condensed liquid is said to be distilled.

The liquid to be distilled is placed in a vessel, called a retort,

generally made of glass or metal, the vapors arising from the boiling liquid pass into a tube, called the worm, which is kept cold by running water, and the condensed vapors flow from the worm into a suitable receiver. The whole apparatus is called a still. If water is treated this way, the condensed steam is pure water, while the salts and other non-volatile substances remain behind in the retort.

Sublimation.—A few solid bodies, when heated, readily change into vapor before they melt, and consequently pass directly over from the solid to the gaseous state. This process, including the subsequent condensation as solid, is called sublimation. Solid iodine, for instance, is such a body, and when heated, passes directly into a gaseous state, and on cooling condenses again to a solid.

Moisture of the Air.

The air, when in contact with water, absorbs some of it, and vapor of water is, therefore, always present in the atmosphere. The warmer the air is, the more moisture can it hold. In a tropical climate the air may become almost saturated with water during the rainy season. But, on the other hand, the air can be hot and still very dry, if it rests over a surface devoid of water, for instance, the great desert region of Sahara.

If warm and moist air is gradually cooled down it finally reaches a temperature at which it can no longer hold all the moisture, but deposits some of it in liquid form as dew, fog or clouds, as the case may be. The less moisture the air contains, the further must it be cooled before this precipitation of water takes place, i. e., the lower is the *dew point.*

The per cent of moisture in the air, as compared with the per cent the air is capable of holding at the same temperature, is termed the *relative humidity* of the air.

Hygrometer.—There are various methods of finding the amount of moisture in the air, and the instruments used for this purpose are called hygrometers.

One is based upon the fact that water will evaporate faster in dry air than in moist. This instrument consists of two thermometers, one of which indicates the actual temperature of the air. The bulb of the other thermometer is covered with muslin, which is kept wet with water. The evaporation produces cold, and

the thermometer soon sinks below the actual temperature of the air. When it comes to rest the temperature is noted, the difference in temperature between the two thermometers is taken, and the amount of moisture in the air found from tables made for this purpose. This instrument is known as the *wet-bulb hygrometer.*

Another common hygrometer is the *hair hygrometer,* in which the variations of moisture in the air are indicated by a hair, previously freed from oily matter, and which stretches when moist and contracts when dry, the movements being transferred to an indicator.

Thermodynamics.

Thermodynamics, or the mechanical theory of heat, is the science of heat in its relation to other forms of energy. As the subject is very complicated, it will be only briefly treated in this book and only those parts taken up which are necessary for the conception of other chapters in the book.

Since heat is a form of energy and can be converted into work, there must be a certain relation between heat units and units of work, or foot-pounds.

The British Thermal Unit, the heat that will raise the temperature of one pound of water one degree Fahrenheit, can do 778 foot-pounds of work. The *mechanical equivalent* of one B. T. U. is, therefore, said to be 778 foot-pounds. This is the *First Law of Thermodynamics,* which states simply that heat and work are mutually convertible in the above proportion. However, it must not be interpreted to mean that whenever we have a gas or liquid containing heat we can simply take the heat away from it and change it all into useful work.

The *Second Law of Thermodynamics* has been variously expressed by different authors. The following is the substance of it: In transforming heat energy into mechanical work it is necessary to have, besides the heat source of temperature, $T_1°$, a refrigerator (or condenser) of a lower temperature. When the working medium (steam, air, gas, etc.) receives a quantity of heat, Q, from the heat source, it will be able, under ideal conditions, to transform only part of this, $\frac{T_1 - T_2}{T_1} \times Q$, into mechanical work. For application of this law, see the chapter "Steam Engines."

Isothermal Changes.—By isothermal expansion or com-

pression is meant one in which the temperature remains constant during the operation. It thus follows Boyle's Law, which states that when the temperature of a gas is constant, the product of pressure and volume remains constant, i. e., $p \times v = p_1 \times v_1$. (See also page 54 and chapter "Refrigeration").

Adiabatic Changes.—By adiabatic expansion and compression is meant one in which the heat remains constant, in other words, no heat is given to or taken from the gas. The pressure-volume relation for a gas in an adiabatic change is given by formula (see page 56 and chapter "Refrigeration").

Electricity and Magnetism

General.

Electricity is the name given to an invisible agent acting as the cause of a large variety of phenomena; such as attraction and repulsion between electrified bodies, heating of a wire through which an electric current is flowing, chemical decomposition of certain substances in solution, etc.

The exact nature of electricity is not yet definitely known, although the effects produced by it, the methods of controlling it and the laws governing its action are becoming more and more understood.

Electrical phenomena may be divided into three groups, namely: *static electricity,* or electricity at rest, which appears as a charge upon bodies; *dynamic* or *current electricity,* or electricity in motion; and *magnetism.*

Electricity is produced by various means, such as *friction, chemical action, heat,* and *induction.*

If a dry glass rod is rubbed with dry silk, the rod acquires the power of attracting light bodies, like bits of paper, straw, etc., and again repelling them. The pieces of paper, after being repelled by the glass rod, are attracted by the silk which electrified the glass rod.

This is explained by assuming that there are two kinds of electrification, the charge on the glass rod being called the *positive* charge and the one on the silk the *negative* charge, and by assuming that charges of the same kind repel, and of opposite kind attract, each other.

The bits of paper, after being in contact with the glass rod, receive a positive charge themselves, and are then repelled from the rod, but the silk, being negatively charged, will then attract them.

One kind of electricity is never produced alone, but equal amounts of positive and negative electricity appear at the same time.

Conductors and Insulators.

Conductors, generally speaking, are substances in which electricity can readily flow from one part to another.

Insulators, or non-conductors, are substances that do not conduct electricity readily, offering considerable *resistance* to its passage. If one end of a glass rod is rubbed with silk it will be electrified, but the other end will not have any electrical charge, or very little of it.

All substances allow some electricity to pass, but there is no substance that does not offer any resistance, so that there is neither a perfect conductor nor a perfect insulator.

Partial conductors are substances of intermediate conductivity, which are neither good conductors nor good insulators.

The following is a list of conductors, partial conductors and insulators given in approximate order of their conductivity:

1.	Metals	8.	Dry wood	15.	Shellac
2.	Charcoal	9.	Marble	16.	Vulcanite
3.	Graphite	10.	Porcelain	17.	Mica
4.	Acids	11.	Paper	18.	Paraffin
5.	Salty solutions	12.	Silk	19.	Glass
6.	Pure water	13.	Rubber	20.	Dry air
7.	Cotton	14.	Gutta percha		

Of these, the metals are classed as good conductors, numbers 2 to 5 fair conductors, 6 to 9 partial conductors, the rest as non-conductors or insulators.

The following table by Matthiessen gives the relative conductivity of various metals at 32° F., silver, which is the best conductor, being taken at 100.

Silver	100.00	Iron (soft)	16.80
Copper (hard)	99.95	Tin	12.36
Gold (hard)	77.90	Lead	8.32
Aluminum	54.00	German Silver	7.50
Zinc (pressed)	29.02	Antimony	4.62
Cadmium	23.72	Mercury (pure)	1.60
Platinum (soft)	18.00	Bismuth	1.24

As copper is an excellent conductor and comparatively cheap, it is used almost universally for conducting electricity from one point to another.

The conductivity of most metals decreases with a rise of temperature, while that of carbon increases.

Electrostatic Induction.

Though the non-conducting substances do not transfer electricity, they permit an electrical influence to be exerted through them. They are, therefore, called *di-electrics*. If, for instance, an electrified body is near to, but not in contact with, an insulated conductor, the electrified body will attract a certain quantity of electricity of an opposite kind to the side near to it and repel the similar kind of electricity to the remote side. Such electrical action is called *induction*.

Electrical Potential.

If an insulated body is charged with electricity and surrounded by a non-conductor, like dry air, the electricity remains in the charged body, in the same manner that the heat given to a body would remain if we could surround the body by a substance that did not conduct heat.

If, on the other hand, the charged body is connected by a conductor with the earth, the electricity will flow from the body through the conductor to the earth. Similarly, if we connect the warm, isolated body by a conductor of heat with a colder body, the temperature of the warm body will sink; heat will flow over to the body of a lower temperature.

When electricity flows from one body to another, the body from which it flows is said to have a higher potential.

The *electrical potential* of a point is, then, the state of that point with reference to its capacity of giving or taking electricity. Potential may be compared to liquid level. Water will flow from higher to lower level in two connected vessels until the level is the same in both.

If two points of different potentials are connected by an electrical conductor, electricity will flow from the point of higher potential as long as a difference is maintained between the two points. Potential may also be compared to the pressure of steam, and the potential is, so to say, the electrical pressure.

Since the term potential is relative, it is convenient to have a standard of reference, the potential of which may be considered 0. The earth is usually assumed to be at zero potential; that is, if a body be charged with positive electricity, the electricity will flow from the body to the earth when connected, and if a body be charged with negative electricity it will flow from the earth to the body.

Condensers.

A condenser, also called *accumulator,* is an apparatus for condensing or accumulating large quantities of electricity upon comparatively small surfaces.

If a conductor, in the form of a plate, is connected with a source of electricity, a certain amount of electricity will flow over to the conductor, until the potential is the same. If, then, another metal plate, connected with the earth, be brought very near to, but not in contact with, the first conductor, the electricity of the first plate will act by induction on the second plate, attracting a certain amount of opposite electricity on the side nearest to it, and expel electricity of the same kind through the conductor to the earth.

The opposite electricity of the second plate will act binding on the electricity of the first plate, and therefore allow another charge of electricity to flow over from the source. Such an arrangement of parallel plates separated by an insulator is called a condenser.

The *Leyden jar* is such a condenser, in which the insulator is a glass jar and the metal plates are tinfoil covers on the inside and outside of the glass jar.

Voltaic Cells.

If a strip of zinc and one of copper, each with a copper wire soldered to its upper end, are dipped into dilute sulphuric acid, both wires become electrified. The wire connected with the copper strip is positively charged, and the wire joined to the zinc negatively. When the two wires are brought in contact, the electricity will flow from copper to zinc continuously, and such a discharge is called an *electric current.* The arrangement of two solids (zinc and copper) in a liquid that acts on the zinc is called a *voltaic cell.*

The two metals are called *elements,* and one of them must be acted upon by the liquid more than the other. The metal acted

upon by the liquid is called the electro-positive element, the other the electro-negative.

The *electrical circuit* is the path along which the electricity flows; it comprises the liquid (called the electrolyte), the elements, and the wire or other conductor, which connects the elements.

The free ends of the wires are called *electrodes* or terminals; the electrode of the negative element is called the *anode,* or positive electrode, and the electrode of the positive element is called the *cathode,* or negative electrode.

In a voltaic cell, the electricity is produced by the chemical combination between the zinc and the acid. The acid dissolves the zinc, and the chemical energy passes into electrical energy when the circuit is closed, in a manner similar to the change of the energy of steam.

Two or more cells connected are called a *battery,* though a single cell is often popularly called a battery.

We distinguish between *primary batteries* and *secondary batteries* or *storage batteries.* Primary batteries are divided into *open circuit* cells, used for intermittent work, and *closed circuit* cells, used for continuous work. Storage batteries are described in the chapter "Electrical Engineering."

Polarization.—A great disadvantage of the simple voltaic cell is that the current strength rapidly decreases when the cell is in action. The chemical action, while generating the current, liberates hydrogen, which gas then collects on the surface of the copper plate and reduces the electromotive force of the cell. This collection of gas with the resultant weakening of current, is called *polarization.*

Depolarizers are substances used in batteries to overcome this collection of gas by absorbing the hydrogen as soon as it is formed.

Amalgamation.—In order to prevent the dissolving chemical action of the acid or solution upon the zinc plate when the cell is inactive, the zinc is coated or amalgamated with mercury. To do this, the zinc is first cleaned by being dipped in weak acid, then rinsed, and then mercury rubbed on its surface. The mercury adheres as a paste-like alloy, and the surplus is wiped or brushed off. In some styles of cells, this amalgamation is prevented from

wearing off or disappearing, by addition of bisulphate of mercury to the solution or acid used.

Kinds of Primary Cells.

There are various kinds of cells in use, differing mainly in the method used for overcoming polarization. Some of these are described below:

Leclanche Cell.—This cell, illustrated by Figure 34, consists of a glass jar containing a solution of ammonium chloride (also called sal ammoniac or battery salt), in which a zinc rod is placed. In the jar is also placed a porous earthenware cup containing a carbon plate surrounded or packed in black oxide of manga-

Figure 34. Leclanché Cell.

Figure 35. Smee Cell.

nese and powdered carbon. The oxide gradually gives off oxygen and thus prevents polarization. The porous cup does not prevent the passage of the current, but protects the zinc from the action of the oxide.

If this style of cell is used for a long period, the strength of the current weakens, but quickly regains its power if the current is discontinued. For this reason it is well suited for purposes where the current is used intermittently, that is for a short time only, as for bells, gas lighting, telephones, etc.

Carbon Cells.—There are various cells now in general use which are of simpler construction than the Leclanche, and have the same action. Such a cell contains a zinc rod, but has no porous cup, carbon being directly in the sal ammoniac solution. The

carbon has different forms, in some batteries it consists of a number of round carbon rods, in others the carbon has the shape of a slotted cylinder, etc. In all of these forms the zinc rod is in the center, being insulated from the carbon by a porcelain thimble and rubber washer.

Smee Cell.—This cell, illustrated in Figure 35, consists of a silver plate, coated with finely divided platinum, the plate being suspended between two zinc plates and the whole immersed in dilute sulphuric acid. Carbon plates are sometimes used instead of silver. Polarization is prevented by the rough surface of the platinum coating.

Figure 36. Bichromate Cell.

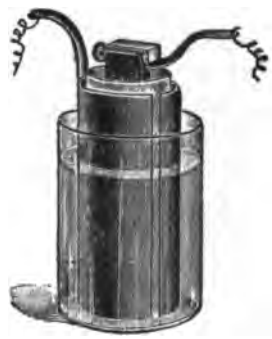


Figure 37. Grove Cell.

Bichromate Cell.—In this cell (Figure 36), a zinc plate is suspended between two carbon plates connected at their tops. The liquid consists of a solution of bichromate of potash and sulphuric acid. The bichromate of potash unites with the hydrogen, thus preventing polarization. The zinc (unless well amalgamated) is quickly attacked, even when the cell is not generating current, and should, therefore, not be left in the solution. By the rod A the zinc plate may be raised above the solution. Bisulphate of mercury is added to this solution for the purpose of keeping the zinc plate amalgamated.

Grove and Bunsen Cells.—These cells (Figure 37), consist of a zinc cylinder inside of which is placed a porous earthenware cup containing strong nitric acid and also a strip of platinum (in the Grove cell) or a carbon plate or rod (in the

Bunsen cell). The space between the porous cup and the jar is filled with weak sulphuric acid. The nitric acid in the porous cup prevents polarization.

Gravity Cell.—This cell (Figure 38), is a modification of the *Daniell cell,* and is generally used in telegraph work. Two liquids are used, one heavier than the other. The heavier one consists of a saturated solution of blue vitriol (copper sulphate), which remains at the bottom and prevents polarization, while the lighter one, a dilute solution of zinc sulphate, remains above. The liquids are thus kept apart by *gravity,* the porous cup used in other batteries being dispensed with. In the lower solution a copper plate is placed, while in the upper solution a zinc plate is used.

Figure 38. Gravity Cell.

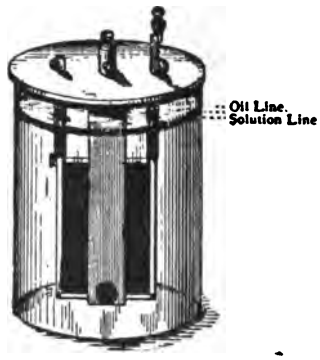


Figure 39. Edison-Lalande Cell.

The two solutions will mix or diffuse gradually when the cell is not in use, but will remain separated when the cell is working; due to the fact that then the copper sulphate solution is changed into zinc sulphate solution so rapidly that there is no time for mixing to take place. The strength of the copper sulphate solution is kept up by placing crystals of copper sulphate at the bottom of the jar or cell, which dissolve as the copper sulphate solution weakens.

Edison-Lalande Cell.—This cell (Figure 39) is of the same type as the Leclanche cell, as it has a single fluid and solid depolarizer. It is, however, much better adapted for work on *closed circut* than the Leclanche cell. The pole plates in the Edison-Lalande cell consist of a copper oxide plate suspended from the cover of the jar between two zinc plates, the latter prevented from coming into contact with the copper oxide by means

of buttons of vulcanite. The liquid is a solution of caustic potash. When the circuit is closed, the current decomposes the water of the solution. Oxygen is deposited on the zinc plates, but immediately combines with zinc and potassium, forming a double salt, which is dissolved as quickly as it is formed. The hydrogen resulting from the decomposition unites with the oxygen of the copper oxide, forming water, and reducing some copper oxide to metallic copper. The cell lasts as long as the copper oxide, during which time an uninterrupted current may be obtained. When all the copper oxide has been reduced to metallic copper, the action ceases. In order to prevent contact with the air, the solution of this cell should be covered with about ¼ inch of paraffin oil.

Dry Cells.— These cells are replacing all others, wherever possible, because they are light, do not freeze, and are not subject to breakage or the spilling of any corroding liquids. This type of cell consists of a zinc cup which forms the outside of the cell and at the same time one of the poles or electrodes. Inside is placed the carbon plate surrounded by the depolarizing and current generating chemicals. The top is sealed with pitch, asphalt or other non-conducting substances.

In using dry cells, care should be taken that they do not touch each other. The reason for this is that the top of the zinc cylinder often flares outwards, so that this flare, together with the protruding binding post, may touch the next zinc or binding post and cause a running down of the cells. It should be remembered that the outer zinc cylinder is one of the plates or electrodes of the cell. Taking the cell out of its pasteboard covering and giving it a coat of shellac varnish, or, still better, painting this pasteboard cover itself with shellac or melted hot paraffin or beeswax, will greatly prolong the life of the dry cell, as this treatment lessens the short-circuiting effect of moist air during damp weather. If the cells are placed on an iron beam or girder they are also likely to quickly run down, unless several layers of pasteboard (preferably shellacked or paraffined) are placed under them.

Grouping of Battery Cells.—In order to obtain stronger currents, two or more cells are connected to form a battery. This is done in three different ways: (1) in series, (2) in parallel, (3) series and parallel combined. For description of this, with calculations, see page 86.

Electromotive Force of Cells.

Bunsen .....1.75 to 1.95 volts	Grove Cell .....1.75 to 1.95 volts
Daniell .....0.98 to 1.08 volts	Leclanche .....1.4 to 1.6 volts
Dry Cell .....1.4 volts	Smee Cell .....0.65 volt
Edison-Lalande .....0.7 volt	

Electrical Quantities and Units.

Electrical Resistance.—As water cannot flow through a pipe without encountering resistance, caused by friction, so also does the conductor through which electricity flows offer resistance to the electric current.

The *unit of resistance* is called the *ohm,* and is the resistance offered to a steady electric current by a column of mercury of 14.4521 grams mass at 32 °F., with a constant cross-sectional area and of a length of 106.3 centimeters. It is approximately the resistance of 1,000 feet of copper wire, 0.1 inch in diameter (No. 10 B. & S. gauge).

The resistance of a conductor is directly proportional to its length, is inversely proportional to its cross-sectional area, and depends upon the specific resistance of the material of which it is made.

Specific resistance of a conductor is its comparative or relative resistance, as compared with that of another conductor taken as a standard. This standard is taken as the resistance of annealed silver.

Current Strength.—The strength of an electric current is measured by its effects, and usually by its chemical effects. The *unit of current strength* is called the *ampere,* and is the current that will deposit by electrolysis from a solution of silver nitrate, under certain fixed conditions, 0.001118 gram of silver per second.

The apparatus used for measuring the strength of an electric current is called an ammeter or amperemeter.

Quantity of Electricity.—The ampere expresses the rate of flow of electricity. The *unit of quantity of electricity* is the *coulomb.* One coulomb is the quantity of electricity that flows in 1 second past any point in the circuit when the current strength is 1 ampere; or, 1 coulomb is the quantity of electricity that is delivered by an ampere current in 1 second. It is therefore often called *ampere-second.*

Since the coulomb is a small quantity, a larger unit, called an *ampere-hour,* is often used. An ampere-hour is the quantity

of electricity delivered by an ampere current in 1 hour, and is, therefore, the same as 3,600 coulombs.

It should be noticed that a coulomb is a certain amount of electricity, and is independent of time, whereas an ampere is a flow of electricity at the rate of 1 coulomb per second. A current of 1 ampere will give 1 coulomb per second; a current of 10 amperes will deliver 10 coulombs per second.

Electromotive Force.—The force that causes the electricity to flow through a circuit is called the electromotive force (E. M. F.), and is sometimes spoken of as the electric pressure. The *unit of E. M. F.* is the *volt*, and is based upon the units of current strength and resistance, and is the E. M. F. that will force a current of 1 ampere through the resistance of 1 ohm. It does work only when it moves electricity.

In a battery, the E. M. F. is caused by the chemical action of the solution upon the plates, and depends upon the nature of the plates and solution, but not upon the size of, or distance between, the plates. In a dynamo, the E. M. F. is caused by a number of wires revolving in a magnetic field, and depends upon the number of wires, the speed, and the strength of the magnetic field.

Electrical Work.—The *unit of electrical work* is the *joule*, or the *volt-coulomb*, and it represents the work done by the electrical energy when 1 coulomb of electricity is moved between two points of a conductor whose difference of potential is 1 volt.

As the coulomb is the quantity of electricity delivered by an ampere current in 1 second, it can also be said that the joule is the amount of work done by an ampere current flowing for 1 second under a pressure of 1 volt. The volt-coulomb, or the joule, is less than a foot-pound.

$$1 \text{ joule} = 0.7375 \text{ foot-pound.}$$

$$1 \text{ foot-pound} = \frac{1}{0.7375} = 1.356 \text{ joules.}$$

Electrical Power.—The *unit of electrical power* is the *watt*. It is the rate of doing work when one ampere passes through a wire at one volt pressure, or it may be stated as work done at the rate of one joule per second. Since watts = volts $\times$ amperes, the watt is sometimes called *volt-ampere*.

Since one joule = 0.7375 foot-pound, one watt = 0.7375 foot-pound per second.

One horse-power is equal to 33,000 foot-pounds per minute, or $\frac{33,000}{60}$, or 550 foot-pounds per second. One horse-power is, therefore, $\frac{550}{0.7375}$ or approximately 746 watts, this being called an *electrical horse-power.*

A still larger unit of electrical power is the ***kilowatt,*** which equals 1,000 watts.

One kilowatt is then equal to $\frac{1,000}{746} = 1.34$ horse-power.

A *watt-hour* is equal to one watt exerted for an hour, which is equal to 3,600 watt-seconds. The *kilowatt-hour is* 1,000 watts per hour, and is the measure generally used in figuring electric light and power currents.

Ohm's Law.

The strength of a current flowing between any two points of a wire is directly proportional to the electromotive force in the wire (difference between its potentials) and inversely proportional to the resistance between these two points.

$$I = \frac{E}{R}$$

I = current strength in amperes.
E = electromotive force in volts.
R = resistance in ohms.

From this it follows that $E = IR$, and $R = \frac{E}{I}$.

Ohm's law forms the basis for all calculations in electrical engineering (see page 84).

Electrical Calculations.

Circular Mils.—A *mil* is the unit of length used in measuring the diameter of a wire, and equals one one-thousandth (0.001) of an inch.

The *circular mil* is the unit of area in measuring the cross-section of a wire or other conductor. It is equal to the area of a circle of one mil diameter.

A *mil-foot* is a wire one foot long and one circular mil in area. To find the area in circular mils of a round wire when its diameter in inches is known, change the diameter into mils and square the number.

Example.—What is the area in circular mils of a wire ¼ inch in diameter?

Solution.—¼ inch = 0.250 inch = 250 mils, and 250^2 = 62,500 circular mils (C. M.).

To find the diameter of a wire when the area in circular mils is known, extract the square root of the circular mil area.

Example.—What is the diameter of a wire, if the area in circular mils is 7,225?

Solution.—$\sqrt{7,225} = 85$ mils, or diameter = 0.085 inch.

Resistance.—As stated on page 78, the resistance depends upon the cross-sectional area, the length of wire, and the material used in the wire.

The resistance of metals increases with the temperature, and that of carbon decreases. The resistance of 1 mil-foot, that is, one foot of wire of 0.001 inch diameter, of various metals, at a temperature of 68° F. is as follows:

Silver	9.84	Zinc	36.69
Copper (at 75° F.)	10.79	Iron	63.35
Aluminium	17.21	Mercury	586.24

These numbers are called the K values of the metals.

The value for copper at other temperatures may be found approximately by adding 0.22% of its value for each degree F. increase in temperature, and deducting this much for each degree decrease.

The resistance of any round metal wire may be found as follows:

$$R = \frac{K \times l}{d^2}$$

R = resistance in ohms.

K = resistance in ohms of 1 mil-foot of wire.

l = length in feet.

d = diameter in mils.

Example.—What is the resistance of 800 feet of copper wire 1/16 inch in diameter at a temperature of 75° F.?

Solution.—The diameter of the wire is 1/16 inch or 0.0625 inch, which equals 62.5 mils. From the table, K = 10.79. Hence,

$$R = \frac{10.79 \times 800}{62.5 \times 62.5} = 2.21 \text{ ohms.}$$

Current Strength and Quantity of Electricity.—Below are given various calculations of current strength and quantity.

Example 1.—An electric current has deposited 3.818 grams of silver on the negative plate of a voltameter in 45 minutes. What was the average strength of current?

Solution.—A current of 1 ampere deposits 0.001118 gram of silver in 1 second; hence

$$I = \frac{3.818}{0.001118 \times 60 \times 45} = \frac{3.818}{3.0186} = 1.26 \text{ amperes.}$$

Example 2.—How many coulombs of electricity were required in the previous example to deposit the silver?

Solution.—Coulombs = amperes × seconds = 1.26 × 60 × 45 = 3,402 coulombs.

Example 3.—An incandescent lamp takes a current of ½ ampere to burn properly. What is the quantity of electricity consumed during 5 hours?

Solution.—Coulombs = amperes × seconds, or

$$\text{Coulombs} = \tfrac{1}{2} \times 3{,}600 \times 5 = 9{,}000.$$

Example 4.—What is the average current strength in a lamp circuit, if 29,000 coulombs are consumed in 4 hours?

$$\textit{Solution.}\text{—Amperes} = \frac{\text{Coulombs}}{\text{Seconds}} = \frac{29{,}000}{3{,}600 \times 4} = 2.01.$$

Example 5.—An electric cell has a capacity of 63 ampere-hours. How many hours can it maintain a current of 3 amperes?

$$\textit{Solution.}\text{—Hours} = \frac{\text{Ampere Hours}}{\text{Amperes}} = \frac{63}{3} = 21 \text{ hours.}$$

Electrical Work.—On page 79, it was shown that one joule = 0.7375 foot-pound, or that one foot-pound = 1.356 joules.

Example.—A current of 25 amperes is maintained through a series of lamps for 4 hours by a pressure of 110 volts; how much electrical work has been performed?

Solution.—Joules = volts × amperes × seconds,

$$= 110 \times 25 \times 60 \times 60 \times 4,$$

$$= 39{,}600{,}000.$$

If resistance is given instead of volts, the volts are found from Ohm's law:

$$I = \frac{E}{R}, \text{ or } E = I \times R; \text{ volts} = \text{amperes} \times \text{ohms},$$

and the previous formula, joules = volts × amperes × seconds, becomes joules = (amperes)^2 × ohms × seconds.

Example.—A current of 5 amperes is sent for 1 hour through an arc lamp of 4 ohms hot resistance; how much work has been done?

Solution.—Joules = $5^2 \times 4 \times 3{,}600 = 25 \times 4 \times 3{,}600 =$ 360,000.

If volts and ohms are given, amperes are found by dividing volts by ohms, $I = \frac{E}{R}$, or amperes $= \frac{\text{volts}}{\text{ohms}}$. The formula for joules then becomes:

$$\text{Joules} = \text{volts} \times \text{amperes} \times \text{seconds},$$
$$= \text{volts} \times \frac{\text{volts}}{\text{ohms}} \times \text{seconds},$$
$$= \frac{(\text{volts})^2}{\text{ohms}} \times \text{seconds}.$$

Example.—A current is sent through a resistance of 2 ohms by a pressure of 30 volts for 10 hours; what is work done?

Solution.—Joules $= \frac{(30)^2}{2} \times 3{,}600 \times 10 = \frac{900 \times 36{,}000}{2} =$ 16,200,000.

Electrical Power Calculations.—Power is the time-rate of doing work, or power $= \frac{\text{work}}{\text{time}}$.

$$\text{Watts} = \frac{\text{joules}}{\text{seconds}} \text{ or } = \text{volt-amperes}.$$

Example 1.—One hundred incandescent lamps, each with 220 ohms hot resistance, are connected in parallel to a 110-volt circuit. How many watts are consumed?

Solution.—The current in each lamp is:

$$I = \frac{110}{220} = \tfrac{1}{2} \text{ ampere}.$$

Watts per lamp $= 110 \times \frac{1}{2} = 55$, and in 100 lamps 55×100, or 5,500 watts.

Example 2.—A 500-watt motor requires 10 amperes; what voltage is needed?

Solution.—Watts = volts × amperes.

$$\text{Volts} = \frac{\text{watts}}{\text{amperes}} = \frac{500}{10} = 50.$$

Example 3.—What horse-power is developed by a dynamo giving 100 amperes and 110 volts?

Solution.—1 horse-power = 746 watts.

Watts = volts × amperes = 110 × 100 = 11,000.

$$\text{Horse-power} = \frac{11{,}000}{746} = 14.7.$$

Example 4.—A generator produces 1,000 amperes and 120 volts; what is its capacity in kilowatts?

Solution.—1 kilowatt = 1,000 watts.

Watts = volt-amperes = 120 × 1,000 = 120,000.

$$\text{Kilowatts} = \frac{120{,}000}{1{,}000} = 120.$$

The Resistance of a Cell.—The total resistance of a cell is divided into external resistance and internal resistance. The *external resistance* is the resistance of the conductor connecting the two plates outside of the cells; the *internal resistance* is the resistance of the liquid between the plates.

A good cell should have a small internal resistance.

Example 1.—A Grenet cell has an E. M. F. of 2.02 volts, the internal resistance is 0.2 ohm, the external resistance is 3 ohms; what will be the strength of current?

Solution.—$I = \frac{E}{R}$, or $\text{amperes} = \frac{\text{volts}}{\text{ohms}} = \frac{2.02}{3+0.2} = \frac{2.02}{3.2} = 0.631.$

Example 2.—What must be the E. M. F. of a cell to send a 0.75 ampere current through a resistance of 1.5 ohms?

Solution.—E = I R = 0.75 × 1.5 = 1.125 volts.

Example 3.—A current of 0.25 ampere is maintained by an E. M. F. of 2 volts; what is the resistance of the circuit?

Solution.—$R = \frac{E}{I} = \frac{2}{0.25} = 8$ ohms.

When we speak of the E. M. F. of a cell, we mean the force that drives the current through the total resistance of the circuit, the internal resistance as well as the external.

If, for instance, a cell of 2 volts has an internal resistance of 0.6 ohm, and is connected with a coil of wire having a resistance of 0.4 ohm (for instance, the coil of the electro-magnet of an electric bell), the current through the coil is not found by

dividing the 2 volts by the 0.4 ohm of the coil, which would give a current of $\frac{2}{0.4}$, or 5 amperes, but by dividing the volts by the total resistance, which is 0.6 + 0.4, or 1 ohm. The current through the coil, as well as through the whole circuit is therefore $2 \div 1$, or 2 amperes.

If we connect the two plates of the cell by a short piece of heavy copper wire, the resistance of which is so small that it can be neglected, then the total resistance would be the internal resistance of the cell alone, or 0.6 ohm, and the current would be $I = \frac{2}{0.6} = 3.33$ amperes.

When the coil is inserted, the E. M. F. of the cell remains the same, but some of the force is spent in driving the current through the coil. The current strength will be less.

If we have steam of 80 pounds pressure in a boiler, and allow the steam to flow out through a 1-inch pipe, and also through a ¼-inch pipe, the pressure is the same in both, but the amount of steam per unit of time is less through the ¼-inch pipe.

Ohm's law applies, however, equally well to a portion of a circuit as to the total circuit, but instead of the total E. M. F., only that part must be used which overcomes the resistance of the portion of the circuit considered.

If we take the previous example, the total E. M. F. of the cell was 2 volts, the internal resistance 0.6 ohm, and the external resistance, that is, the resistance of the coil, was 0.4 ohm, consequently, the current $I = \frac{2}{0.6 + 0.4}$, or 2 amperes.

The current, whatever it may be, is always the same in all parts of the circuit, and is, therefore, 2 amperes in the coil. Since the resistance of the coil is 0.4 ohm, the E. M. F. between the coil terminals is found thus:

$$I = \frac{E}{R}, \text{ or } E = IR = 2 \times 0.4, \text{ or } 0.8 \text{ volt.}$$

Of the total E. M. F. of 2 volts, 0.8 volt is used to overcome the resistance of the coil, and $2 - 0.8$, or 1.2 volts, to overcome the internal resistance.

We distinguish, therefore, between the total E. M. F. and

the difference of potential, or *volts drop,* between the terminals of a portion of the circuit.

Example.—An incandescent lamp has a hot resistance of 220 ohms, and is connected to an electric light main, across which is maintained a potential difference of 110 volts. What current will flow through the lamp?

Solution.—$I = \frac{E}{R} = \frac{110}{220} = \frac{1}{2}$ ampere.

Connection in Series and in Parallel.

When a number of cells are connected so that the positive pole of one cell is joined to the negative pole of the next cell, and so on, the cells are said to be connected in *series.* The E. M. F. of the first cell is added to the E. M. F. of the second, and the total E. M. F. of the battery is the E. M. F. of each cell multiplied by the number of cells. The total internal resistance is the sum of the internal resistance of all the cells.

If the internal resistance of each cell is called r (ohms), and the external resistance R, the E. M. F. of each cell is called E; then if one cell only is used, the current is

$$I = \frac{E}{r + R}$$

If 4 cells are used, all equal, and the same external resistance, R, then current is

$$I = \frac{4E}{4r + R}$$

When all the positive poles are connected together on one side, and all the negative poles on the other, the connection is said to be in *parallel,* or *multiple.* If we have, for instance, 4 cells connected in such a manner, the 4 cells form really 1 cell with 4 times larger plate surface. In such case the total E. M. F. is the same as the E. M. F. of one cell, but the internal resistance is only one-fourth. If we call the internal resistance of each cell r, the external resistance R and the E. M. F. of the cell E, then the battery of 4 cells gives a current

$$I = \frac{E}{\frac{r}{4} + R.}$$

Example.—A Leclanche cell has an E. M. F. of 1.4 volts, and

an internal resistance of 0.4 ohm. What is the current, using one cell, if the outside resistance (electric bell) is 0.1 ohm?

Solution.—$I = \frac{1.4}{0.4 + 0.1} = \frac{1.4}{0.5} = 2.8$ amperes.

Example 2.—What is the current, using 4 such cells, connected in series, with the same external resistance?

Solution.—$I = \frac{4 \times 1.4}{4 \times 0.4 + 0.1} = \frac{5.6}{1.6 + 0.1} = \frac{5.6}{1.7} = 3.3$ amperes.

Example 3.—What is the current in the above problem, if connection is in parallel?

Solution.—$I = \frac{1.4}{\frac{0.4}{4} + 0.1} = \frac{1.4}{0.1 + 0.1} = \frac{1.4}{0.2} = 7$ amperes.

When the outside resistance is small (0.1 ohm) it is then better to use the connection in parallel. But suppose that the outside resistance is considerable, for instance, 100 ohms instead of 0.1 ohm. Under such circumstances the 4 cells arranged in series would give a current:

$$I = \frac{4 \times 1.4}{4 \times 0.4 + 100} \text{ or } \frac{5.6}{1.6 + 100} \text{ or } \frac{5.6}{101.6} \text{ or } 0.05 \text{ ampere.}$$

Arranged in parallel the current would be

$$I = \frac{1.4}{\frac{0.4}{4} + 100} \text{ or } \frac{1.4}{0.1 + 100} \text{ or } \frac{1.4}{100.1} \text{ or } 0.01 \text{ ampere.}$$

The arrangement in series in this case gives 5 times stronger current.

If two or more cells are connected in series and two or more such combinations are connected in parallel, we have the *parallel-series* connection. With this method of connecting cells, increased voltage and decreased resistance are obtained.

Conductance.

The conductance of a circuit is the reciprocal of its resistance. If the resistance of a conductor is 4 ohms, the conductance is ¼ *mhos* (ohms spelled backward), and if resistance is R ohms, conductance is $\frac{1}{R}$ mhos.

If two or more resistances are added in series (one after another), the joint resistance (JR) is the sum of the resistances.

If two resistances of 40 and 160 ohms are added one after another in a circuit, the joint resistance, or JR = 40 + 160, or 200 ohms. The current is the same in both, but the drop of potential in each spool is proportional to its resistance. If, for instance, the current is 0.01 ampere, then the potential difference between the terminals of the spool with 40 ohms resistance is $I \times R = 0.01 \times 40$, or 0.4 volt, and in the other spool 0.01×160, or 1.6 volts.

Resistances in Parallel.

If two or more equal resistances are joined in parallel, their joint resistance is the resistance of one of them divided by their number, or $JR = \frac{R}{n}$, where R is the resistance of one branch and n their number.

Example 1.—Four incandescent lamps are connected in parallel, and each lamp has a hot resistance of 212 ohms. What is the total resistance of the lamp circuit?

Solution.—$JR = \frac{212}{4} = 53$ ohms.

Example 2.—The joint resistance of 10 lamps in parallel is 22 ohms. What is the resistance of each lamp?

Solution.—$R = 22 \times 10 = 220$ ohms.

If the resistances are unequal, for instance, if one resistance of 40 ohms and another of 60 ohms are added in parallel, we make use of the conductance, by saying that the conductance of the two resistances is equal to the sum of the conductances in each.

The amount of electricity that flows through the resistance of 60 ohms is less than that which flows through the 40 ohms, which is also expressed when we say that the conductances of the two branches are 1/60 and 1/40 of a mho. The joint conductance is, then:

$$1/60 + 1/40 = \frac{40 + 60}{60 \times 40} = \frac{100}{2{,}400} = 1/24 \text{ mho.}$$

The joint resistance therefore is 24 ohms. Generally, if R and R_1 indicate the resistance in each branch, then their joint resistance equals $\frac{R + R_1}{R \times R_1}$.

The following example shows how to find the strength of current in each branch of a divided circuit:

Example.—A current of 50 amperes is passed through a circuit, which in a certain part is divided into three branches in parallel, and the branches have resistances of 2, 5 and 8 ohms, respectively. How many amperes will each branch take?

Solution.—Since the resistances are 2, 5 and 8 ohms, the respective conductances are 1/2, 1/5 and 1/8 mho. This means that the currents in the three branches are to each other as 1/2 : 1/5 : 1/8 or as 20/40 : 8/40 : 5/40 or as 20 : 8 : 5. Since $20 + 8 + 5 = 33$, then the current that flows through the 2 ohms resistance is 20/33 of the total 50 amperes, or,

$$\frac{20 \times 50}{33} = \frac{1{,}000}{33} = 30.3 \text{ amperes.}$$

The current through the 5 ohms resistance is 8/33 of 50, or $\frac{400}{33}$ = 12.1 amperes.. The current through the 8 ohms resistance is 5/33 of 50, or $\frac{250}{33}$ or 7.6 amperes.

Shunts.

If it is desirable to send only a part of a current through a circuit, the other part of the current may be sent around that portion of the circuit on a by-path, which is then called a shunt. If, for instance, a current is too large for a galvanometer, only a small portion of it may be passed through the galvanometer by passing the remainder through a wire connected across the galvanometer terminals. The galvanometer is then said to be shunted.

The ratio of the total current to the part that passes through the galvanometer is called the multiplying power of the shunt, and is found by dividing the sum of the galvanometer resistance and the shunt resistance by the shunt resistance.

Electrical Measurements.

Measurements of Current.—The voltameter and the tangent galvanometer are used for standardizing the commercial instrument, and in laboratories for measuring the strength of current. The instrument used in practice is called an amperemeter or *ammeter,* and it shows directly the number of amperes of current flowing through a circuit.

Since an ammeter must take the total current, it must always be connected in series between the generator and the apparatus

which receives the current. It should, therefore, have a low resistance, so as not to absorb much of the current, and the needle should come to rest quickly, or should be *dead beat.*

Measurement of Pressure.—The instruments used to measure the electrical pressure are called *voltmeters,* and are constructed on the same principles as the ammeters, only the coils are made of very fine wire suitable for the small current that has to go through them. Like the ammeter, the voltmeter also measures the current flowing through it, but the calibration is in terms of pressure. Voltmeters are never connected in series, but across the line of which the potential difference is wanted, or in parallel with the conductor between the ends of which the voltage is to be measured.

Example 1.—A dynamo maintains 110 volts at the lamps, and 22 lamps are connected, each of 220 ohms resistance. What current does the ammeter indicate?

Solution.—Joint resistance $= \frac{220}{22} = 10$ ohms, and $I = \frac{110}{10} = 11$ amperes.

Example 2.—What current goes through the voltmeter if its resistance is 110,000 ohms?

Solution.—$I = \frac{E}{R} = \frac{110}{110{,}000} = \frac{1}{1{,}000}$ ampere.

Example 3.—Suppose the ammeter has a resistance of 0.1 ohm and is placed across the circuit, what current would then pass through the ammeter?

Solution.—$I = \frac{E}{R} = \frac{110}{0.1} = 1{,}100$ amperes.

Measurement of Resistance.—There are various methods for determining the resistance of a conductor. A very simple method is by employing a battery voltmeter and ammeter. The unknown resistance is connected to the battery or source of current, the ammeter is inserted in series, and the voltmeter across the ends of the resistance. The current should not be so strong that heating of the conductor occurs. Readings of the two instruments should be taken simultaneously, and since $R = \frac{E}{I}$, the resistance is easily calculated. If for instance, the ammeter

shows 6 amperes, and the voltmeter 18 volts, the resistance is then $R = \frac{18}{6} = 3$ ohms.

Effects of the Electric Current.

Chemical Effects.—An electric current produces chemical changes. If a current is passed through water, the current separates the water into its two component elements, hydrogen and oxygen. If led through a solution of nitrate of silver in water, it deposits silver on the negative electrode (zinc pole).

Electro-chemical action is called *electrolysis.*

Magnetic Effects.—For description of magnetism caused by electric current, see under "Magnetism," page 92.

Heating Effect.—When a current is passed through a wire, the resistance causes heating, that is, electricity is changed into heat.

Joule's Law states that the number of heat units generated in a conductor in a given time is directly proportional to the resistance, to the square of the current strength and to the time. Thus, if the resistance were doubled in the same current, the amount of heat generated would be doubled; but if the current strength were doubled in the same resistance, the heat generated would be four times as much.

This heating effect upon a poor conductor is applied practically in blasting, heating of street cars, electric welding, electric cooking, fuses, and principally in electric lighting by means of incandescent lamps. In the latter the current passes through a carbon filament or other wire possessing a high resistance, which becomes heated to incandescence and gives off light. In the other applications, the same principle underlies their operation, insofar as the heat is generated by the current passing through nickel-wire resistance coils, plates, etc. In electrical welding, the ends of the iron offer the resistance and become red hot.

Magnetism.

Substances which have the property of attracting iron are called magnets, and this power of attraction is called magnetism.

Kinds of Magnets.—There are two kinds of magnets, *natural* and *artificial* magnets. Natural magnets are found in nature in the form of magnetic oxide of iron, also called lode-

stone. Artificial magnets are made when a natural magnet is rubbed against a bar of steel, by which the steel acquires the same magnetic property, only to a much larger extent, for which reason artificial magnets are used instead of natural magnets.

Magnets are also classified as permanent and temporary.

Permanent magnets are such as retain their magnetic properties indefinitely, or lose them only very gradually. These are of steel.

Temporary magnets.—When a piece of soft iron is touched by or brought close to a permanent magnet it also becomes magnetic and remains so as long as it is close to or in contact with this permanent magnet, but loses its magnetic properties again when removed from it.

Magnetic Poles.—The ends of a magnet are called its

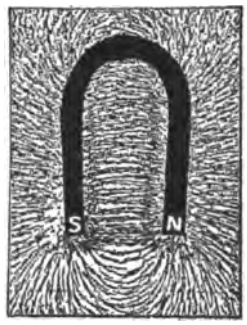


Figure 40. Lines of Magnetic Force.

poles; the one which points north is usually called the north pole (the north seeking pole); the other is called the south pole.

If the north pole of a magnet is brought near a suspended magnet, it will repel the north pole and attract the south pole of the suspended magnet, showing that like poles *repel* and unlike poles *attract* each other.

Magnetism by Electricity.—If a few turns of an insulated wire are wrapped around a steel bar, and a current of electricity is passed through the wire, the steel bar will be permanently magnetized. If an iron bar be substituted for the steel bar, the iron remains a magnet only as long as the current flows.

The force exerted by a magnet is called the magnetic force; it emanates in all directions from the magnet.

Magnetic Field.—The space around a magnet which is permeated by the magnetic force is called the magnetic field of force, or simply the magnetic field. The directions of the magnetic forces in the space around the magnet are called the magnetic lines or *lines of force*. The magnetic lines issue from the north pole and enter through the south pole.

Magnetic Induction.—When a piece of soft iron is brought into a magnetic field it also becomes a magnet. This action is called magnetic induction, and takes place whether the iron is in contact with, or separated from, the magnet by a non-magnetic body, such as wood, glass, air, etc. However, a piece of heavy iron placed between the magnet and the iron acts as a magnetic screen, and does not allow the magnetic lines to pass through.

Magnetic Circuits.—A simple magnetic circuit is made up entirely of a magnetic substance having a uniform cross-sectional area, for instance, an iron ring. A compound magnetic circuit is made up of magnetic and non-magnetic substances, for instance, the iron ring with a section removed (air gap).

If a magnetic body is free to move under the influence of a magnetic field it will place itself so as to let through the greatest possible number of lines of force from the field.

To concentrate and direct the lines of force the poles of a magnet are connected with a pole piece of soft iron fastened to the poles.

In a dynamo, pole pieces of soft iron are fastened to the end of the magnet, and between them is a magnetic body near to, but not touching, the pole pieces, and free to rotate. This movable piece is called an armature.

Electromagnetism.

Electromagnetism is the magnetism produced by the electric current around a conductor. A wire carrying a current is surrounded by what are called magnetic whirls, being a succession of circular and parallel lines of force. The directions of these lines of force are such that if one look along the wire in the direction in which the current flows the whirls are clockwise (from left to right). Air offers more resistance to the passage of magnetic lines of force than does iron or steel. The capability of a substance for conducting magnetic lines of force is called

its *permeability*. The permeability of iron may exceed that of air by 2,000 times.

The magnetizing force of a coil of wire is directly proportional to the strength of the current and the number of times it flows around the coil.

4 amperes and 25 turns = 100 ampere turns.
2 amperes and 50 turns = 100 ampere turns.

It is the magneto-motive force of the ampere turns which drives the magnetic lines through the magnetic circuit.

The resistance of a magnetic circuit is called *reluctance*, and this reluctance must be overcome before induction is produced.

The number of lines of force per square inch of the circuit is called induction per square inch, and corresponds to the current in an electric circuit, and

$$\text{Induction} = \frac{\text{Magnetizing Force}}{\text{Reluctance}}.$$

Electromagnetic Induction.—If a wire forming a closed circuit is moved across a magnetic field, a current of electricity is produced in the wire. Currents produced this way are called induction currents, and the phenomenon is termed electromagnetic induction, and upon this principle is based the operation of induction coils, telephones, dynamos, etc.

It is immaterial whether the wire is moved and the magnet stationary or reversely. Opposite poles of a magnet produce currents in opposite directions, and an electromagnet has the same effect as an ordinary magnet. The current is produced, not at the expense of the magnetic force, but by the force moving the wire.

The dynamo is a machine which converts mechanical energy into electrical by means of electromagnetic induction. It consists of two metallic circuits, one of iron or steel, for the magnetic force, and one of copper for the electric current.

By moving one of these circuits in a proper way, the mechanical energy is transformed into electrical, and if electrical energy is supplied, mechanical energy may be produced (electric motor).

For details see chapter "Electrical Engineering."

Chemistry

This chapter has been given a place in the book to serve as a reference for the user. It is not intended to be a thorough treatise on chemistry, and includes as far as possible only those substances which may directly or indirectly be of interest to the engineer.

The Non-Metallic Elements and Their Compounds.

Oxygen (O) is a colorless, odorless and tasteless gas. It is the most abundant of all elements. It occurs free in the air, but mechanically mixed with about four times its volume of nitrogen. In combination with other elements it occurs in enormous quantities. Thus, for instance, it constitutes about 88.9% by weight of water, large percentages of the earth's crust, and of plants and animals. It combines readily with almost all elements. All substances which burn in air, burn with increased brilliancy in oxygen gas. Many substances usually considered incombustible, like iron, will burn in oxygen if their temperature is raised sufficiently to initiate the combustion. It is very difficultly liquefied, its critical temperature being — 182°F. and its critical pressure 765 pounds. Liquid oxygen is a mobile transparent liquid, possessing a faint blue color. At atmospheric pressure, it boils at —296° F., at which temperature it has a specific gravity of 1.138.

The action of chemically combining with oxygen is called *oxidation*. Depending upon conditions, it may be slow, as when iron rusts in the atmosphere; or rapid, as when a substance takes fire and burns in the air. The action opposite to oxidation, i. e., withdrawal of chemically combined oxygen, is known as *reduction*.

When an electric discharge is passed through oxygen, a portion of it is changed into an allotropic, i. e., modified form,

Properties of the Most Important Elements.

Element.	Symbol.	Character.*	Atomic Weight.†	Specific Gravity Water 1.00.	Specific Gravity Hydrogen 1.00.	Melting Point in Degrees F.
Aluminium	Al.	Metal	27.1	2.58—2.69	.....	about 1,160
Antimony	Sb.	Metal	120.2	6.70—6.86	.....	810
Argon	Ar.	Gas	39.88	.....	19.95	.....
Arsenic	As.	Metal	74.96	5.39—5.96	.....	835—855
Barium	Ba.	Metal	137.37	4.0	.....	over 2,600
Bismuth	Bi.	Metal	208.0	9.70—9.83	.....	515—518
Boron	Bo.	Amorphous solid	11.0	2.45	.....	.....
Bromine	Br.	Mobile liquid	79.92	3.188	.....	19
Cadmium	Cd.	Metal	112.40	8.56—8.70	.....	590—610
Calcium	Ca.	Metal	40.07	1.52—1.56	.....	about 1,500
Carbon	C.	Amorphous solid	12.00	2.2 (graphite)	.....	.....
		Crystalline solid		3.53 (diamond)		
Chlorine	Cl.	Gas	35.46	.....	35.68	.....
Chromium	Cr.	Metal	52.0	6.81	.....	about 3,300
Cobalt	Co.	Metal	58.97	8.80	.....	2,785
Copper	Cu.	Metal	63.57	8.81—8.94	.....	1,830—2,200
Fluorine	F.	Gas	19.0	.....	18.85	.....
Gold	Au.	Metal	197.2	19.37—19.55	.....	1,942—1,947
Helium	He.	Gas	3.99	.....	2.16	.....
Hydrogen	H.	Gas	1.008	.....	1.000	.....
Iodine	I.	Crystalline solid	126.92	4.948	.....	238
Iridium	Ir.	Metal	193.1	22.38	.....	4,500
Iron	Fe.	Metal	55.84	7.5—7.84	.....	2,800—2,900
Lead	Pb.	Metal	207.10	11.35—11.44	.....	622
Lithium	Li.	Metal	6.94	0.59	.....	356
Magnesium	Mg.	Metal	24.32	1.69—1.75	.....	1,175

Element.	Symbol.	Character.	Atomic Weight.	Specific Gravity Water 1.00.	Specific Gravity Hydrogen 1.00.	Melting Point in Degrees F.
Manganese	Mn.	Metal	54.93	7.4—8.0	.....	about 3,300
Mercury	Hg.	Liquid metal	200.6	13.59—13.60	.....	—38
Molybdenum	Mo.	Metal	96.0	9.01	.....	about 3,300
Nickel	Ni.	Metal	58.68	8.27—8.93	.....	2,550—2,900
Nitrogen	N.	Gas	14.01	.....	13.975	.....
Osmium	Os.	Metal	190.9	21.30—22.48	.....	4,500
Oxygen	O.	Gas	16.00	.....	15.848	.....
Phosphorus	P.	Waxy solid	31.04	1.836	.....	112
Platinum	Pt.	Metal	195.2	20.85—21.71	.....	3,100—3,250
Potassium	K.	Metal	39.10	0.865	.....	144
Radium	Ra.	Metal	226.4	.....	.....	.....
Selenium	Se.	Crystalline solid	79.2	4.48	.....	423
Silicon	Si.	Crystalline solid	28.3	2.34	.....	about 2,600
Silver	Ag.	Metal	107.88	10.51—10.61	.....	1,761—1,764
Sodium	Na.	Metal	23.00	0.973	.....	204
Strontium	Sr.	Metal	87.63	2.54	.....	about 1,500
Sulphur	S.	Crystalline solid	32.07	2.06	.....	235
Tantalum	Ta.	Metal	181.5	16.64	.....	4,100—4,200
Tellurium	Te.	Crystalline solid	127.5	5.93—6.39	.....	850
Thorium	Th.	Metal	232.4	11.0	.....	.....
Tin	Sn.	Metal	119.0	7.29	.....	446
Titanium	Ti.	Amorphous solid	48.1	4.87	.....	.....
Tungsten	W.	Metal	184.0	19.13	.....	about 5,000
Uranium	U.	Metal	238.5	18.7	.....	about 2,900
Vanadium	V.	Metal	51.0	5.5	.....	3,050
Zinc	Zn.	Metal	65.37	6.91—7.19	.....	780
Zirconium	Zr.	Metal	90.6	4.25	.....	.....

* Under normal conditions.
† According to Annual Report of International Committee on Atomic Weights, 1913.

called ozone. *Ozone* is a pale blue gas of a peculiar and characteristic smell, somewhat suggestive of dilute chlorine. It is a powerful oxidizing gas which rapidly destroys organic matter. When inhaled, it irritates the mucous membrane and rapidly induces headache. Ozone liquefies easily to an intensely blue liquid.

Hydrogen (H) is a colorless, odorless and tasteless gas. It does not occur in the free state on the earth except in minute quantities (it is present, free, in the solar atmosphere, in certain fixed stars, in nebulae and comets). Combined with other elements it is present in many minerals, in natural gas and petroleum, in all organized matter, both animal and vegetable. Combined with oxygen it constitutes about 11.1% of water. It is thus one of the most widely distributed and abundant of elements. Hydrogen is the lightest of all elements, and for this reason is accepted as the unit for atomic weight. It burns in air or oxygen with a pale blue flame, the product of the combustion being water. When mixed with air (or oxygen) in certain proportions and touched by a burning taper, it explodes with great violence. Hydrogen is not poisonous, but undiluted it cannot be respired for any length of time, as it does not support life. Hydrogen is very difficultly liquefied, the critical temperature being — 389°F., and its critical pressure 294 pounds. Liquid hydrogen is colorless, and it is the lightest liquid known, its specific gravity being 0.07. It boils at — 425° F., and becomes solid at about — 430° F.

Hydrogen combines with oxygen to form two oxides, namely, hydrogen monoxide and hydrogen dioxide. Hydrogen monoxide, commonly called *water* (H_2O) is a compound, which when pure is composed of 88.86% oxygen and 11.14% hydrogen, by weight; or, of one volume of oxygen gas and two volumes of hydrogen gas. Pure water is tasteless, odorless and colorless, except when viewed through great depths, when it is pale blue. (The color of great spans of water, like oceans and lakes, is due to suspended particles and reflections). In common with most liquids it is a very poor conductor of heat and electricity and is but slightly compressible. In the presence of very small quantities of either bases or acids, it is decomposed by the electric current into its elementary constituents.

Water is never found pure in nature on account of its great solvent action on other substances. Besides forming the oceans, lakes, rivers, wells, clouds, etc., it is present in many crystalline salts as *water of crystallization,* and in all animal and vegetable life. Pure water is the accepted unit for comparing specific gravities of liquids and solids.

Hydrogen dioxide or peroxide is a very unstable compound. In its pure form it is a syrup of specific gravity 1.5. Commercially it is sold in dilute aqueous solutions of from 3 to 25 per cent. Its solution in water reacts feebly acid. It is a powerful oxidizing agent, and is employed as a bactericide in surgery. At ordinary temperatures, it decomposes slowly into water and oxygen.

Nitrogen (N) is a colorless, odorless and tasteless gas. It is found in the free state in the atmosphere, of which it constitutes about four-fifths by volume. It is essential to the life of plants and animals, and is a constituent of all albuminous substances. Nitrogen does not readily combine with other elements, it does not burn, nor does it support combustion. It is not poisonous, but is incapable of supporting respiration. Nitrogen liquefies with comparative ease. Its critical temperature is —231° F., and its critical pressure 500 pounds. It forms a transparent colorless liquid, boiling at —315° F., at which temperature it has a specific gravity of 0.804, and becomes solid at —355° F.

Atmosphere or *air* is the name commonly applied to the mixture of gases which surrounds the earth. When freed from the impurities, which are always present in varying amounts, the air from any part of the globe is found to be a very uniform mixture. The average composition by volume, expressed in parts per million is as follows:

Pure air	Oxygen	206,594.0
	Nitrogen	771,160.0
	Rare gases: Argon	7,881.0
	Helium, Xenon, Neon, and Krypton	19.0
Impurities present in varying quantities.	Carbon dioxide	336.0
	Ozone	1.5
	Aqueous vapor	14,000.0
	Nitric acid	8.0
	Ammonia	0.5

The various impurities found in the air, have not only a very important effect on the economy of nature, but are also a source of corrosion of the many things built by man, which are exposed to their influence. Rain-water is always contaminated by the absorption of these impurities, both the natural and those which are characteristic of manufacturing plants, for which reason the air is always pure after rain. Pure air is frequently used as a unit for comparing the specific gravity of gases.

Nitrogen combines with oxygen forming five oxides, of which the most important are *nitrogen trioxide* (N_2O_3), which is the anhydride of nitrous acid; and *nitrogen pentoxide* (N_2O_5), the anhydride of nitric acid.

Nitric acid (HNO_3) is a colorless and very pungent liquid of strong acid properties. It is intensely corrosive, and powerfully oxidizing on account of the readiness with which it parts with oxygen. The nitric acid of commerce is, even in its purest form, mixed with water; the pure acid of sp. gr. 1.41 contains 67.5% nitric acid and 32.5% water. It dissolves most of the metals, forming the corresponding nitrates. Nitric acid is without action on the so-called noble metals: gold, platinum, iridium, etc. When hydrochloric acid is added to nitric acid (usually in the proportion of 3 to 1), a mixture is obtained, which on account of its dissolving action on gold was given the name *aqua regia* by the alchemists. The solvent action on the rare metals is due to the free chlorine evolved.

Carbon (C) occurs native in three allotropic forms, viz.: diamond, graphite and coal. The *diamond* in its purest form is colorless and crystalline. It is the hardest substance known, and is unaffected by all acids. The diamond is combustible and burns with great brilliancy when heated in oxygen, forming carbon dioxide.

Graphite is found native in large quantities and in various degrees of purity. Extreme variations in the composition of graphite are shown in the following table:

Specific Gravity	2.105 — 2.586
Volatile Substances	1.1% — 7.3%
Carbon	40.0% — 96.0%
Mineral Matter	4.8% — 46.2%

Some specimens of Siberian and Ceylon graphite have a carbon content as high as 96%, whereas those from the continent frequently fall below 50%.

Graphite is a lustrous grayish black solid, soft and smooth to the touch. It is usually found as compact laminated masses, more seldom as prismatic crystals. It is a good conductor of heat and electricity. When strongly heated in oxygen it burns with brilliancy, forming carbon dioxide, and leaving an ash consisting of silica, alumina, and ferric oxide. Graphite is employed in the manufacture of lead pencils, dry lubrication, paints, crucibles, and for foundry facings. It has lately met with success as a boiler compound, but as its action is purely mechanical, it cannot be used for boiler waters containing the chlorides of calcium and magnesium or free acids (see also "Boiler Compounds").

Coal, which is the only form of amorphous carbon occurring to any extent in nature, is always associated with varying amounts of hydrogen, nitrogen, sulphur and non-volatile mineral substances. The chemical composition of coal varies considerably, as is shown in the following table of elementary composition of anthracite and bituminous coals (see also "Combustion"):

	C	H	O	N	S	Ash
	%	%	%	%	%	%
Anthracite	82.5—92.0	1.5—3.3	0.5— 3.0	0.2—0.9	0.1—10.0	3.0— 6.0
Bituminous	75.5—86.0	4.2—5.3	4.0—12.9	0.6—1.9	0.2— 4.0	3.0—20.0

Other forms of amorphous carbon, all of which are the products of manufacture are: Lampblack, gas carbon, wood charcoal and animal charcoal.

Lampblack is a very soft and flocculent substance which is produced by burning substances rich in carbon, in a limited supply of air. The soot is deposited in chambers on suspended coarse cloths. It is always accompanied by hydrogen, but is readily purified leaving a pure amorphous carbon.

Gas Carbon is obtained by the destructive distillation of coal. It is extremely hard and is a good conductor of electricity. It is employed in the manufacture of carbons for arc-lamps.

Wood Charcoal is a very impure form of carbon, and is obtained almost exclusively as the residue from the destructive distillation of wood. Its properties vary considerably, depending upon the wood used and the method of manufacture. It is a poor conductor of electricity.

Animal Charcoal is obtained as a residue from the destructive distillation of bones. It contains the least carbon of all ordinary forms. The average composition is: carbon 10%; inorganic matter, chiefly calcium and magnesium phosphates 90%.

Wood charcoal, and animal charcoal to a marked degree, possess the property of absorbing large volumes of gases, and of removing coloring matters from solutions. The absorbed gas is held mechanically by the charcoal in a highly condensed form on the surface of the porous mass. Fetid gases, produced by the putrefaction of organic matter, on absorption, are oxidized by the oxygen previously absorbed by the charcoal from the air.

Carbon combines with oxygen to form two oxides: carbon monoxide and carbon dioxide.

Carbon monoxide (CO) is a colorless, tasteless and odorless gas. It is the product of the incomplete combustion of coal. It burns in the air with a pale blue flame forming carbon dioxide. At higher temperatures it acts as a powerful reducing agent, which fact has been made use of in a number of metallurgical processes. Carbon monoxide is an extremely poisonous gas, even small quantities when present in the air giving rise to headache and dizziness, and if inhaled for some time, unconsciousness and death quickly follow.

Carbon dioxide (CO_2) is a colorless gas with a faint and pleasantly pungent smell and slight acid taste. It is frequently found as a constituent in mineral waters, and is present in small quantities in the air and in all natural waters. It is the main source from which plants derive their carbon. In combination with other oxides, especially that of calcium as calcium carbonate, it forms a great portion of the earth's crust. Carbon dioxide does not burn and is incapable of supporting combustion. It is moderately soluble in water at atmospheric pressure, its solubility being considerably increased by an increase

in pressure, as is illustrated in the so-called "aerated" or carbonated waters. When carbon dioxide is dissolved in cold water and especially under increased pressure, it gradually combines with an equivalent quantity of water, forming *carbonic acid* (H_2CO_3), an unstable compound, which on exposure to the atmosphere at a slightly increased temperature, slowly decomposes into carbon dioxide and water, the former escaping into the air. On account of this combining power, the carbon dioxide escapes more freely from its solution immediately after being absorbed, than after storing for some time. Carbon dioxide is readily condensed at 32° F. under a pressure of 492 pounds to a colorless and extremely mobile liquid. Its critical temperature is 88° F., and its critical pressure is 1,088 pounds. At atmospheric pressure, liquid carbon dioxide boils at — 109° F.

The specific heat of CO_2 gas at constant pressure is 0.2169, at constant volume, 0.1714. The specific heat of the liquid is 0.79; the specific gravity varies from about 0.6 to 1.2, according to temperature.

Carborundum is a silicide of carbon, formed by the action of carbon on sand at high temperature. It is an important abrasive.

The compounds of carbon with other elements are very numerous, and form by far the largest chapter of the science of chemistry. Until about the middle of the nineteenth century the name *organic* was applied to this chapter, which included only substances derived from the products of animal and vegetable life. The original meaning of this term, however, has long ago disappeared, and has been replaced by the more correct *Chemistry of the Carbon Compounds.* According to their behavior, these compounds are divided into numerous groups.

The most important of these groups are the hydrocarbons or paraffin series, alcohols, aldehydes, ketones, acids, carbohydrates, fatty acids, albumens, alkaloids.

The Halogens is a term applied to the four elements fluorine, chlorine, bromine and iodine, on account of the great similarity of their sodium salts to ordinary sea-salt. (Halogen is a Greek word, meaning sea-salt producer). Each of the four elements unites with hydrogen, forming respectively hydrofluoric acid (HF), hydrochloric acid (HCl), hydrobromic acid (HBr), and hydriodic acid (HI), all of which are color-

less gases which fume strongly in the air. They are all soluble in water and react strongly acid.

Chlorine (Cl) is a greenish yellow gas with a strong disagreeable smell. In its pure state it is highly poisonous and causes death; even when much diluted with air it acts upon the mucous membrane causing irritation and inflammation. Chlorine is a powerful antiseptic, and possesses strong bleaching properties.

Hydrochloric acid (HCl) is a colorless pungent gas, which, when cooled, condenses to a colorless liquid. It is extremely soluble in water. The hydrochloric acid of commerce has a sp. gr. of 1.21, and contains about 43% of the acid and 57% of water. It dissolves a large number of metals and their oxides, forming the corresponding chlorides.

Sulphur (S) is a pale yellow crystalline solid, insoluble in water. It dissolves readily in carbon disulphide, from which it again crystallizes upon evaporation. It is a poor conductor of heat and a non-conductor of electricity. Sulphur is found native chiefly in volcanic districts where it is mined.

Sulphur combines with oxygen to form four oxides; the most important of which are sulphur dioxide, and sulphur trioxide.

Sulphur dioxide or *sulphurous anhydride* (SO_2) is a colorless gas having a strong suffocating odor. It is produced when sulphur is heated and burned in the air. It is very soluble in water, combining with it to form *sulphurous acid* (H_2SO_3). This acid acts as a reducing agent, being slowly oxidized by free oxygen, and rapidly by oxidizing agents, to sulphuric acid. It is a bleaching agent, possesses marked preserving properties, and has to some extent been used as an antiseptic. Sulphur dioxide is easily liquefied; its critical temperature being 247° F., and critical pressure 1,142 pounds. Liquid sulphur dioxide is a colorless mobile liquid which boils at 14° F., at atmospheric pressure, and solidifies to a transparent ice-like mass when cooled to — 105° F. It is used as a refrigerant.

Sulphur trioxide or *sulphuric anhydride* (SO_3) is an exceedingly volatile liquid which crystallizes when cooled below 59° F. It fumes strongly when exposed to air, on account of

combining with the water vapor, forming minute drops of sulphuric acid.

Sulphuric acid or *oil of vitriol* (H_2SO_4) is a colorless heavy oily liquid. It absorbs moisture from the atmosphere and withdraws water from many substances when mixed with them, even when the water is chemically combined. It attacks organic matter, finally decomposing it, in many cases even at low temperatures. Sulphuric acid combines with almost every element, forming the corresponding sulphate.

Sulphuretted hydrogen (H_2S) is a colorless gas, having a sickly somewhat sweet taste and an extremely offensive odor. It is highly poisonous when inhaled in the pure state, and even when diluted with air it frequently produces headache and dizziness. Hydrogen sulphide escapes from fissures in the earth in volcanic districts, and is a constituent of many mineral springs. It is also a product of decomposing sewage, and this accounts for the strong "sulphur smell" of some contaminated wells. The flue gases from the incomplete combustion of coals rich in pyrites also contain this gas.

Phosphorus (P) is an unstable translucent very pale yellow and wax-like solid. It is volatile at ordinary temperatures and is readily inflammable, taking fire in the air when heated to 93° F., and burning with a bright white light. When exposed to the air it emits a faintly luminous pale greenish light and gives off white fumes which are poisonous and have an unpleasant smell. Water is practically without action upon phosphorus, for which reason it is always preserved under water. It is soluble in carbon disulphide. It is never found free in nature. Phosphorus combines with oxygen to form five oxides, the most important of which is *phosphorus pentoxide* or *phosphoric anhydride* (P_2O_5), a soft flocculent powder which on sublimation forms extremely deliquescent transparent crystals.

Ortho-phosphoric acid (H_3PO_4), commonly called *phosphoric acid,* is a transparent very deliquescent crystalline solid, melting at 107° F. It is a derivative of phosphorus pentoxide formed by digesting its solution in water.

Silicon (Si) exists in two forms, the amorphous and the crystalline. Amorphous silicon is a dark brown powder, which

burns in the air when heated, forming silicon dioxide. It is insoluble in water and all acids except hydrofluoric acid. In the crystalline form, it exists as brilliant steel gray, very hard needles which do not burn when heated in the air, nor in oxygen. It is insoluble in all acids except a mixture of nitric and hydrofluoric acids. Neither form occurs free in nature. In combination with oxygen, only one oxide is known, namely silicon dioxide.

Silicon dioxide or *silica* (SiO_2) occurs in nature in two forms; the amorphous, a translucent substance having a conchoidal fracture; and as quartz, which in its purest form is found as colorless prismatic crystals. It is a non-conductor of electricity and a poor conductor of heat. It melts with difficulty in the electric arc, to a transparent glass-like mass, which is capable of being formed into various apparatus employed by the chemist and physicist. Silicon dioxide is unacted upon by all acids excepting hydrofluoric acid. In combination with water, silicon dioxide forms weak acids, and as such is found in most natural waters.

Boron (B) is a dark brown amorphous substance. It is unaffected by air at ordinary temperatures, sublimes in the electric arc, and burns on strong ignition, forming the trioxide and nitride of boron. Heated with sulphuric acid it is oxidized to boric acid, and fused with alkaline hydroxides and carbonates, it forms the corresponding borate. In combination with oxygen, only one oxide, the trioxide, is known.

Boric acid or *boracic acid* (H_3BO_3) is obtained by dissolving boron trioxide in water. It forms white pearly laminated crystals, difficultly soluble in cold water, but readily in hot water. It is a weak acid, possessing feeble antiseptic properties.

The Metallic Elements and Their Compounds.

The **Alkali Metals** are a group consisting of the five metals lithium (Li), sodium (Na), potassium (K), rubidium (Ru), and caesium (Cae). All of these elements resemble each other to a marked degree in their chemical and physical properties. They are all soft, silvery white metals, which rapidly tarnish in the air, and decompose water at ordinary temperatures. They are malleable, melt at moderate temperatures and

are comparatively volatile. They are all lighter than water. Their oxides and hydroxides react strongly alkaline. With acids they form a large series of compounds, the most important being those formed by sodium and potassium.

Sodium hydroxide or caustic soda (NaOH) is a white brittle and extremely deliquescent solid, which dissolves readily in water with a considerable evolution of heat. Its solution reacts strongly alkaline and is extremely caustic.

Sodium chloride or *common salt* (Na Cl) forms anhydrous colorless crystals, which melt at about 1,500° F. It is readily soluble in water and alcohol. It is found in enormous quantities in nature; dissolved in sea water, from which it is obtained on a large scale by a system of evaporation; and in large deposits, where it is mined. Salt is a necessary article of food for man and animals. In strong solutions it is extensively used for refrigerating purposes on account of its low freezing point.

Sodium carbonate or *washing soda* ($Na_2CO_3 + 10$ aq.) forms large colorless transparent crystals which effloresce, i. e., lose their water of crystallization on exposure to air, finally falling to a powder. Its solution reacts strongly alkaline and is mildly caustic. *Soda ash* is the technical name for the commercial dry sodium carbonate.

Sodium sulphate or *Glauber salts* ($Na_2SO_4 + 10$ aq.) forms large colorless prisms which effloresce on exposure to the air. The crystals melt in their own water of crystallization at 91° F. It occurs in nature in the anhydrous form as the mineral thenardite and it is a constituent of many natural waters. Its solution acts as a laxative.

Hydrogen disodium phosphate or *phosphate of soda* ($Na_2H PO_4 + 12$ aq.) forms large colorless prisms which effloresce on exposure to the air and melt in their own water of crystallization at 95° F.

Normal sodium orthophosphate or *trisodium phosphate* ($Na_3PO_4 + 12$ aq.) crystallizes in thin colorless prisms which absorb carbon dioxide from the atmosphere, crumbling to a powder, with the formation of sodium carbonate and hydrogen disodium phosphate. Its solution reacts alkaline. Used as a boiler compound, it acts upon the chloride, sulphate and bicarbonate of calcium and magnesium, precipitating these bases as

phosphates and carbonates; the sodium remains in solution as chloride and sulphate.

Sodium silicate or *water glass* is a compound of indefinite composition. It is produced by the fusion of silica with sodium hydroxide. It is used as a binding material for asbestos; in the manufacture of artificial stone; for fireproofing of wood; and as a filler in cheap soaps. It has lately been successfully employed as a boiler compound, the silica reacting with the calcium and magnesium salts, forming the corresponding silicates which are insoluble.

Sodium borate or *borax* ($Na_2B_4O_7 + 10$ aq.) forms transparent crystals which have a very astringent taste. When heated, borax loses its water of crystallization and puffs up to a very porous mass; at red heat it melts, forming a transparent colorless liquid. In the molten state it has the power to combine with metallic oxides. It is extensively used in the manufacture of porcelain and glass enamel.

Potassium hydroxide or *caustic potash* (KOH) closely resembles the corresponding sodium compound, except that it is considerably more energetic in its action.

Ammonia (NH_3) is a compound gas. It is colorless and extremely pungent, having a strong caustic taste. It is unable to support combustion, and in the concentrated form causes suffocation. Ammonia gas is extremely soluble in water, 1 volume of which at 32°F. and atmospheric pressure is capable of absorbing 1,180 volumes of ammonia gas at the same temperature and pressure; the gas is again expelled by boiling. The saturated solution has a specific gravity of 0.882 at 15° C., and contains 35% of ammonia. The solution reacts strongly alkaline. Ammonia gas is easily condensed to a colorless mobile liquid which boils at —27° F. and solidifies to a white crystalline mass at —103° F. Its critical temperature is —266° F., and its critical pressure 1,675 pounds. Ammonia gas is present in the atmosphere in very small quantities. It is produced commercially by the destructive distillation of nitrogenous organic matter; and in nature by the putrefaction of nitrogenous substances. For other properties, see "Refrigeration."

Ammonia, though a compound, possesses the peculiarity that in chemical reactions its molecule combines with another atom

of hydrogen, forming the radical *ammonium* (NH_4) which has elementary properties, taking part in chemical reactions, in which it is transposed unaltered. With acid it forms a series of compounds which resemble those of the alkali metals, but unlike in that they are volatile.

Ammonium carbonate [$(NH_4)_2CO_3$+ aq.] forms large colorless crystals which decompose at 140° F. into ammonia, carbon dioxide and water. It is very unstable, losing ammonia on exposure to air, the bicarbonate of ammonia being formed. The commercial salt is a mixture of ammonium bicarbonate and ammonium carbamate [$(NH_4)HCO_3$+ $NH_4CO_2NH_2$].

Ammonium chloride or *sal ammoniac* (NH_4Cl) forms colorless fernlike crystals which are very soluble in water with absorption of heat, and consequent cooling of the solution. On boiling the aqueous solution, a slight dissociation takes place, ammonia escaping and the solution at the same time becoming slightly acid.

The Alkaline Earths.—This group comprises five elements: Glucinum or beryllium (Gl), magnesium (Mg), calcium (Ca), strontium (Sr), and barium (Ba), none of which is found native. The first two are white metals which retain their luster for a considerable time in the air. The last three have a yellow color and metallic luster, which on exposure to air tarnishes, the metal becoming converted into the oxide. The oxides of magnesium, calcium, strontium, and barium absorb water with the formation of the corresponding hydroxide. With acid they form a large number of salts, the most important being those of magnesium and calcium.

Magnesium oxide (MgO) is a white bulky powder which fuses in the electric arc and in the oxy-hydrogen blowpipe, and on cooling solidifies to a vitreous extremely hard mass. On account of its refractoriness, it is used for a number of metallurgical purposes.

Magnesium chloride ($MgCl_2$+ 6 aq.) forms large colorless deliquescent crystals. When heated, it loses water and finally decomposes into magnesium oxide and hydrochloric acid. In the presence of certain salts, such as ammonium chloride, it can be fused without decomposition. It is present as a natural constituent in sea water, many mineral springs and some well

waters. Such waters cannot be used for boiler feeding on account of the dissociation of the magnesium chloride at boiler temperatures. The judicious use of suitable feed-water treatment can remedy this defect. Its aqueous solution may be used for refrigerating purposes (see chapter "Brine System").

Magnesium sulphate or *Epsom salts* ($MgSO_4 + 7aq.$) crystallizes in colorless prisms which effloresce on exposure to air and become anhydrous when heated to about 390° F. It is very soluble in water, its solution having a bitter taste and strong laxative action. It is present as a natural constituent of many mineral wells, and occurs in nature as the mineral kieserite.

Calcium oxide or *lime* (CaO) is a white amorphous substance, which readily absorbs carbon dioxide and water from the air, forming the carbonate and hydroxide of calcium. When proportionately small amounts of water are added to calcium oxide, the mass rapidly becomes hot, giving off considerable steam, and at the same time it swells up and finally crumbles to a soft powder. This product, technically known as *slaked lime* is *calcium hydroxide* ($Ca(OH)_2$). It is soluble in water; its solution reacts alkaline and absorbs carbon dioxide with the precipitation of calcium carbonate. The mixture of slaked lime with less water than is required for dissolving it, it known as *milk of lime*. *Mortar* is a mixture of slaked lime, water and large quantities of sand; the hardening of which is a very slow process, depending upon the interaction of the carbon dioxide of the atmosphere with the calcium hydroxide.

Calcium carbonate ($CaCO_3$) occurs native in large deposits of limestone, marble, chalk, and a large variety of calcspars. Its solubility in water is about 1/10 grain per gallon. The presence of carbon dioxide in the water, considerably increases its solubility, with the formation of calcium bicarbonate, 41 grains of which are soluble in 1 gallon at 32°F. Under increased carbon dioxide pressure, 1 gallon of water is capable of dissolving as much as 175 grains. By heating the water or by exposing it to the air, the carbon dioxide is expelled, and calcium carbonate is precipitated.

Calcium chloride ($CaCl_2 + 6aq.$) forms large colorless deliquescent crystals which melt at 84° F. in their water of

crystallization. When heated above 390° F., it loses all its water, with the formation of *anhydrous calcium chloride;* a porous mass which is extremely hygroscopic. In this form it is employed as a dessicating agent. At red heat it fuses without decomposition, and on cooling, solidifies to a crystalline and very hygroscopic mass, technically known as *fused calcium chloride.* This is extensively used for refrigerating purposes (see chapter "Brine System").

Bleaching powder or *chloride of lime* is an unstable compound, consisting of calcium hypochlorite combined with calcium chloride. Its chemical composition varies considerably, depending upon its method of manufacture and upon its age. It decomposes slowly, even in a closed vessel. The available chlorine in the freshly prepared compound varies from 30 to 38%. Very dilute acids liberate hypochlorous acid, which in the presence of an excess of dilute hydrochloric acid is decomposed into chlorine and water.

Calcium sulphate ($CaSO_4$) occurs native as the mineral anhydrite and alabaster. It is a constituent of many lake and well waters. Calcium sulphate is obtained as a fine crystalline powder by precipitation, technically known as *precipitated gypsum* ($CaSO_4 + 2aq.$) In this form it contains 20.9% of water, present as water of crystallization. When this compound is dried at 250—265° F., it loses part of its water, and as such, is known as *plaster of Paris* ($2CaSO_4 + aq.$). In this form, when made into a paste with water, it rapidly sets to a hard mass; for which reason it has found extensive use in the arts, such as the making of plaster casts, stucco, artificial stone, etc. When gypsum is heated above 390°F., it becomes anhydrous. Calcium sulphate is but slightly soluble in water, 1 part dissolving in 500 parts of water at 64° F.

Aluminium (Al) is a comparatively dull white metal. It is very ductile, malleable, and highly electropositive. It does not tarnish in the air under normal conditions, but oxidizes when strongly heated. It melts at about 1,290° F. It is but slightly affected by acids, excepting hydrochloric acid, which readily dissolves it with the evolution of hydrogen gas. It is readily soluble in solutions of potassium hydroxide and sodium hydroxide. The presence of impurities renders it much more unstable. Aluminium combines with oxygen to form one oxide.

Aluminium oxide or alumina (Al_2O_3) is an amorphous soft white substance. It is insoluble in water, but dissolves in most acids with the formation of the corresponding salts. It occurs native in a colorless crystalline form as corundum, and, colored by traces of metallic oxides as ruby, sapphire, amethyst, etc. These forms are extremely hard, ranking second only to that of the diamond.

Kaolin or *china clay* is a hydrous aluminium silicate ($H_4Al_2Si_2O_9$) occurring widely distributed in nature, usually considerably mixed with impurities. It is used in the manufacture of porcelain, pottery, tiles, etc.

The *alums* are a group of double salts, the most common of which is potassium alum, generally known as alum. It is a double sulphate of potassium and aluminium ($K_2SO_4 + Al_2(SO_4)_3 + 24aq.$) It crystallizes in characteristic colorless crystals, which effloresce on exposure to air. This efflorescence is not due to a loss of water, but to the absorption of atmospheric ammonia with the formation of a basic salt. It is soluble in water, the amount dissolved increasing considerably with the rise in temperature. The solution has a weak acid reaction and an astringent taste. It melts in its own water of crystallization, which on further application of heat is expelled, until at a low red heat, the salt is changed into a bulky white porous mass, known as *burnt alum*. In this state it dissolves very slowly in water. It is employed as a water purifier and as a mordant.

Antimony (Sb) is a bright white, crystalline and brittle metal. It is a poor conductor of heat and electricity. It melts at 805° F. When heated in the air, it takes fire and burns with a bright light forming antimony trioxide. It is oxidized by nitric acid and is soluble in concentrated hydrochloric and sulphuric acids. It forms two oxides, and in combination with other elements, a large number of compounds. Antimony is found free in nature to but a small extent, its principal ore being the black prismatic mineral stibnite, in which it occurs as trisulphide. Antimony is employed in the manufacture of a number of important alloys.

Bismuth (Bi) is a lustrous, white, extremely brittle metal, with a faint reddish tint. It melts at 515° F. It is a poor

conductor of electricity. It tarnishes very slowly in the air, but when strongly heated, takes fire and burns, forming the trioxide. It is slightly soluble in hydrochloric acid and cold sulphuric acid. Bismuth forms four oxides, and in combination with other elements, a large number of compounds. It is found free in nature, usually as arborescent and granular masses. Other ores of commercial importance are bismuth ochre, in which it occurs as trioxide; and bismuth glance, in which it is present as trisulphide. Bismuth is employed in the manufacture of a number of alloys.

Copper (Cu) is a lustrous reddish brown metal, very ductile and malleable. It is an extremely good conductor of electricity. Copper can be drawn into very fine wire and can be hammered into very thin leaf, but the presence of impurities greatly reduces these properties. It melts at about 2,000° F. When heated to near its melting point it can be reduced to a fine powder. It tarnishes slowly in moist air. Copper is readily soluble in nitric acid, but is unaffected by hydrochloric and sulphuric acids except in the presence of atmospheric oxygen, when dilute acids affect it. Solutions of ammonia also affect it in the presence of air; the oxide which is formed, dissolves, producing a blue solution of ammonium cuprate. With other metals, copper forms a large number of important alloys.

Copper occurs native in enormous quantities, and in combination with other elements is widely distributed. Its principal ores, aside from the native, are: ruby ore, in which it occurs as cuprous oxide; copper glance, copper pyrites and purple or peacock ore, in which it occurs as sulphide; and malachite and azurite, in which it occurs as basic carbonate.

Copper forms two oxides: *Cuprous oxide* (Cu_2O) a dull red amorphous powder, and *cupric oxide* (CuO) a dull black amorphous powder. In combination with other elements, it forms a large number of compounds, the most important of which is copper sulphate.

Copper sulphate or blue vitriol ($CuSO_4 + 5$ aq.) forms large blue transparent crystals, which on heating lose water, being changed into the anhydrous salt, a very pale blue almost white amorphous powder, which is hygroscopic. Copper sulphate finds application in electric batteries, in copper plating,

and as a germicide, especially for destroying the stubborn algae in water reservoirs.

Silver (Ag) is a white lustrous metal. It is unaffected by pure air, but tarnishes readily in the presence of sulphuretted hydrogen. It is the best conductor of heat and electricity of all metals. It is extremely malleable and ductile. It melts at 1,763° F. It dissolves readily in nitric acid, but is insoluble in hydrochloric and in cold sulphuric acid.

Silver occurs native in but very small quantities; its principal ores being sulphides, frequently in combination with antimony, arsenic, and copper. Silver is present in almost all lead ores. Silver forms three oxides, and in a combination with other elements, a large number of compounds, the most important of which are the chloride and the nitrate.

Silver chloride (AgCl) is a very unstable white amorphous powder, which rapidly blackens on exposure to light, absorbing oxygen from the atmosphere. It melts at 860° F. to a clear liquid, and solidifies on cooling to a translucent resinous mass. Silver chloride is insoluble in water, but soluble in ammonium solutions.

Silver nitrate ($AgNO_3$) crystallizes in anhydrous colorless plates, which are comparatively stable. It fuses at 424° F. It is soluble in water, its solution staining the skin and acting as a violent poison.

Gold (Au) is a soft yellow metal, the most malleable and ductile of all metals. It melts at about 1,950° F. It does not tarnish in the air and is unaffected by all acids except selenic acid and aqua regia.

Zinc (Zn) is a bluish-white highly crystalline and brittle metal. When heated in the air it takes fire and burns with a brilliant bluish-white light, forming oxide of zinc. Zinc melts at 780°F. It is unaffected by dry air, but tarnishes slightly under normal atmospheric conditions. The pure metal is practically unacted upon by dilute or strong hydrochloric and sulphuric acids, but in the presence of small quantities of impurities it dissolves readily, with the evolution of hydrogen. Zinc is extensively used for *galvanizing* iron, a process which consists in dipping the iron into a bath of molten zinc. The layer of zinc protects the iron from the action of the atmos-

phere, preventing rusting. With other metals, it forms a number of important alloys. Zinc combines with oxygen to form one oxide; and in combination with other elements, it forms a number of important compounds. The chief sources of zinc are the mineral zincite in which it occurs as oxide, smithsonite as carbonate, and zinc blende as sulphide.

Zinc oxide (ZnO) is a pure white flocculent powder, which turns yellow on heating, becoming white again on subsequent cooling. It is insoluble in water, but dissolves in acids, forming the corresponding zinc salts. Zinc oxide is largely used as pigment in paints.

Zinc chloride ($ZnCl_2 + aq.$) is a soft, white crystalline and very deliquescent solid. It fuses at about 480° F. to a white translucent mass. It is easily soluble in cold water, forming, when concentrated, a thick colorless syrup. Its solution is partly decomposed on heating, a basic salt being precipitated. Solutions of zinc chloride are used for impregnating wood, to prevent the growth of organisms. Used as a soldering solution, it removes metallic oxides from surfaces which are to be soldered.

Cadmium (Cd) is a bright white metal possessing a bluish tint. It is very malleable and melts at about 600° F. It is but slightly affected by dry air, but in moist air it becomes superficially coated with the oxide. On heating the metal to redness, it takes fire and burns, forming a brown colored oxide. It is a constituent of several fusible alloys.

Mercury (Hg) is a bright silver white liquid metal. It vaporizes slightly at ordinary temperatures and is extremely poisonous. At —39° F. it freezes. It tarnishes very slowly under ordinary atmospheric conditions. Mercury is soluble in nitric acid, but insoluble in hydrochloric and sulphuric acids at ordinary temperatures. It alloys with almost all the metals, forming what are known as mercury *amalgams*. Mercury occurs native in small globules disseminated through its ores. The chief ore is cinnabar, in which it occurs as sulphide. Mercury forms two oxides, and in combination with other elements, a large number of compounds.

Mercurous chloride or *calomel* (Hg_2Cl_2) is a white trans-

lucent and fibrous mass. It is insoluble in water. When heated it vaporizes without fusion.

Mercuric chloride or *corrosive sublimate* ($HgCl_2$) crystallizes in long white silky needles. It is soluble in cold water and is a violent poison.

Tin (Sn) is a bright white crystalline metal of low tenacity and great malleability. It melts at 446° F., and volatilizes at red heat. When strongly heated it takes fire and burns, forming stannic oxide. It is insoluble in concentrated nitric acid, but is oxidized by it to metastannic acid. It dissolves in hydrochloric acid forming stannous chloride. Tin is extensively used in the process of *tinning,* which consists in dipping other metals into a bath of molten tin. Tinplate is sheet iron which has been superficially coated in the above manner. Tin forms two oxides, stannic oxide and stannous oxide, and two well marked series of salts derived therefrom. Tin does not occur native, its principal ore being tin stone or cassiterite, in which it occurs as stannic oxide.

Lead (Pb) is a soft bluish-white lustrous metal. It tarnishes readily under ordinary atmospheric conditions. It melts at 619° F. Lead is readily soluble in nitric acid, but practically insoluble in hydrochloric and sulphuric acids. In the absence of air it is unaffected by pure water. The solvent action of water upon lead is greatly increased by the presence of various dissolved substances, such as small quantities of nitrates of calcium and magnesium, ammonium salts, carbon dioxide, etc. Physiologically considered, lead is a cumulative poison of considerable magnitude, for which reason, lead piping should under no condition be employed for conveying potable water. Lead rarely occurs native, the chief source being the mineral galena, in which it occurs as a sulphide; it is also found as a carbonate in the mineral cerussite, as sulphate in anglesite, as basic sulphate in cenarkite, and as a basic chloride in matlockite. Mixed with other metals, lead forms a number of important alloys. Lead combines with oxygen to form five oxides, of which two are used in the arts.

Plumbic oxide, litharge or *massicot* (PbO) is the oxide formed when lead is strongly heated in the air. It is a reddish

yellow powder, very slightly soluble in water, but soluble in acids and molten alkalies.

Triplumbic tetroxide, red lead, or *minium* (Pb_3O_4) is a scarlet crystalline powder. It is soluble in dilute acids with the formation of the corresponding salts. Commercial red lead varies considerably in composition and color, depending upon the method of manufacture. It is mostly used as a pigment.

In combination with other elements, lead forms a large number of salts, the most important of which are the chlorides and the carbonates.

Plumbic chloride or *lead dichloride* ($PbCl_2$) crystallizes in long white lustrous needles which are slightly soluble in water. The basic compound ($Pb_2OCl_2 + aq.$) is prepared by the addition of lime water to a solution of lead chloride and is employed as a white pigment.

Lead carbonate ($PbCO_3$) is a white crystalline powder almost insoluble in water, but more readily soluble in water charged with carbon dioxide. The basic carbonate of lead [$2PbCO_3Pb(OH)_2$] is a heavy white amorphous powder, largely used as a white pigment under the name of *white lead.* Used in paints, it possesses extreme opacity and density (covering power), but has the disadvantage of "turning," i. e., becoming discolored in time due to the action of sulphuretted hydrogen, which is always present in the air.

Vanadium (V) is a white metal. It is one of the rare metals and is extremely difficult to obtain in its metallic state. It dissolves slowly in hydrofluoric acid and in nitric acid, forming a blue solution. It is insoluble in hydrochloric acid. It combines with oxygen to form five oxides, and with other elements to form a large number of compounds. Though vanadium is widely distributed, it occurs only in small quantities; its chief ores being the minerals vanadite, motramite, various clays, iron ores and pitchblende. It is frequently found in Bessemer slag. In the form of ferro-vanadium it is added to steels in the manufacture of vanadium steel.

Chromium (Cr) is a hard steel gray metal. It does not tarnish, but burns when heated in oxygen forming chromic oxide. Chromium liquefies in the electric arc. It is soluble in hydrochloric acid. The metal is non-magnetic. It is found

chiefly in combination with iron in the mineral chromite and with lead in crocoisite. It is not found in the free state in nature. Chromium forms three oxides and three series of compounds: The *chromous salts,* which correspond to chromous oxide (CrO), the *chromic salts,* corresponding to chromic oxide (Cr_2O_3), and the *chromates,* corresponding to chromic trioxide (CrO_3), which is an acid anhydride.

Molybdenum (Mo) forms a gray colored metallic powder. It is malleable and comparatively hard. It is rapidly oxidized on heating. It is soluble in dilute nitric acid and in concentrated sulphuric acid. Molybdenum forms several oxides and a large number of compounds. It occurs in nature chiefly as the minerals molybdenite and wulfenite, also to some extent in iron ores.

Tungsten (W) is a steel gray crystalline metal. It tarnishes very slowly in the air, and burns when heated to redness in the air, forming the trioxide. Tungsten melts at about 5,000° F. It is soluble in aqua regia, nitro-hydrofluoric acid and is slowly attacked by sulphuric acid, hydrochloric acid and hydrofluoric acid. It forms several oxides and a large number of compounds. It occurs chiefly in the minerals wolfram, scheelite and stolzite. Tungsten is employed in the manufacture of filaments for electric lamps, and in steel alloys (see page 133).

Manganese (Mn) is a grayish-white, hard and brittle metal. It oxidizes rapidly on exposure to moist air. It is readily soluble in dilute sulphuric and hydrochloric acids. Manganese forms several oxides and a large number of compounds. It occurs in nature in various minerals, the most important of which are pyrolusite, braunite, manganite and hausmannite.

Iron (Fe) is a white lustrous metal, which rapidly tarnishes in moist air, especially in the presence of carbon dioxide, a hydrated oxide of iron [$Fe_2O_3Fe(OH)_2$] being formed. This action is commonly called *rusting.* At red heat, iron decomposes water into its components. Iron is soluble in dilute acids with evolution of hydrogen and the formation of the corresponding salts. Iron is the most abundant of the heavy metals, occurring in enormous deposits. Its chief ores are: red and brown hematite, magnetic iron ore and specular iron ore, in all

of which the iron is present as oxide; spathic iron ore, in which it is present as carbonate; and pyrites, in which it is present as sulphides.

The physical properties of iron are to a great extent influenced by various impurities. Thus pure iron when magnetized quickly loses this property, whereas steel retains it at ordinary temperatures. Pure iron does not take a temper when heated and suddenly cooled, whereas steel treated in this manner becomes extremely hard and brittle.

For metallurgy of iron and steel, see page 123.

Iron combines with oxygen to form three oxides; ferrous oxide, ferric oxide, and ferroso-ferric oxide.

Ferrous oxide, iron monoxide, or *protoxide of iron* (FeO) is a black amorphous powder which oxidizes readily in the air. It is soluble in dilute acids.

Ferric oxide or *sesquioxide of iron* (Fe_2O_3) is a reddish-brown amorphous powder. It is easily soluble in dilute acids. When produced by the strong ignition of iron salts, it forms steel gray to brilliant black crystalline granules, which pulverize to a reddish brown powder and dissolve with great difficulty in dilute or concentrated acids. *Rouge* is a form of ferric oxide obtained as a palpable powder in the manufacture of Nordhausen sulphuric acid. It is used as a pigment and for polishing.

Ferroso-ferric oxide or *magnetic oxide of iron* (Fe_3O_4) is formed when iron burns in oxygen, and by the action of air on red hot iron. The so-called *hammer scales* or *smithy scales,* a blackish gray porous mass, produced in the forging of iron, consists chiefly of this oxide. It occurs in nature as magnetic iron ore, the richest iron ore known.

In combination with other elements, iron forms two well defined series of salts, the ferrous salts, corresponding to ferrous oxide, and the ferric salts, which correspond with the ferric oxide, besides exhibiting feebly acid properties in the formation of ferrates and ferrites.

Ferrous sulphate or *green vitriol* ($FeSO_4 + 7aq.$) forms large pale green transparent crystals which are very soluble in water. On exposure to air, it effloresces and oxidizes to form a basic ferric sulphate.

Ferrous sulphide (FeS) is a dark yellowish gray metallic

mass. It is decomposed by dilute sulphuric and hydrochloric acids, with the evolution of sulphuretted hydrogen and the formation of the corresponding ferrous salt.

Ferrous carbonate ($FeCO_3$) is a white amorphous powder, which rapidly oxidizes on exposure to air, it is found in some mineral waters (chalybeate), being soluble to some extent in carbonic acid water, from which it is usually precipitated in an oxidized form on exposure to air.

Ferric chloride ($Fe_2Cl_6 + 6aq.$) forms yellow crystals which are extremely deliquescent. It is obtained in the anhydrous form in black scaly crystals which appear red by transmitted light and green by reflected light. The dilute solution is slowly decomposed on exposure to air.

Cobalt (Co) is a bright white, malleable and very tough metal. Like iron and nickel it is attracted by the magnet; but unlike them, it retains this property at higher temperatures. It is unaffected by dry air, except in the finely divided state, when it is rapidly oxidized. It is slowly soluble in dilute acid. Cobalt forms several oxides and a large number of compounds. Cobalt is comparatively scarce, its chief ores are the minerals speiss-cobalt, cobalt glance and cobalt bloom, in all of which it occurs in combination with arsenic. It is frequently found in small quantities together with nickel and with iron in their ores.

Nickel (Ni) is a lustrous grayish white metal with a faint yellow tinge when compared with silver. It is attracted by the magnet, but like iron it loses this property when moderately heated. It is ductile and malleable, yet very hard, and burnishes to a very high polish. Nickel is unaffected by dry air but tarnishes slowly when exposed to ordinary atmospheric conditions, becoming coated with a thin film of oxide. It melts at about 2,730° F. It dissolves slowly in dilute hydrochloric and sulphuric acids with the formation of their corresponding salts. With other metals it forms several important alloys. Nickel forms several oxides and a large number of compounds. Nickel occurs in the free state in meteorites; its chief ores are millerite, nickel blende and nickeliferous pyrrhotines in which it occurs as sulphide; in niccolite it occurs as arsenite, and in garnierite as silicate.

The last of the elements are the group of three rare metals **Osmium** (Os), **Iridium** (Ir), and **Platinum** (Pt). They are all white metals which melt with great difficulty. Osmium is the heaviest substance known. Both osmium and iridium are insoluble in all acids, but osmium dissolves when fused with potassium hydroxide, forming potassium osmiate. Platinum has the lowest melting point of the three metals, melting at about 3,200° F. It does not tarnish in the air and is insoluble in all acids excepting aqua regia. The three metals are found alloyed in nuggets and small grains in alluvial deposits and river sand, platinum forming the larger per cent, though occasional nuggets are richer in either of the other two.

Alloys.

Alloys are intimate mixtures of metals, obtained by melting and subsequent cooling. The process modifies the properties of the individual metals. Thus, for instance, the electrical conductivity of a metal is very much decreased by alloying with even small quantities of other metals; a nickel steel containing 36% of nickel retains an almost constant volume when heated or cooled; chromium in steel hardens it; the tensile strength of aluminium is greatly increased by alloying with a few per cent of magnesia.

Alloys were formerly considered solid solutions, i. e., homogeneous mixtures, but recent research has shown that this is not correct. In the metallographical examination of alloys, which consists in examining a prepared surface of the alloy with medium to high power microscopes, the surface is shown as a complex pattern frequently crystalline, and composed of at least two different materials.

As the composition of alloys vary, some of them considerably, the percentages given in the table, page 122, are only approximate. For more detailed information regarding this matter, the reader is referred to Kent "Mechanical Engineers' Pocket Book."

Fusible Metals.

	Melting point Degrees F.	Bismuth	Cadmium	Lead	Tin
Lipowitz.	140°	50	10	27	13
Wood's..	150—160°	57	15	14	14
Rose's...	175—195°	49	..	27	24
Darcet's.	200°	50	..	25	25
Newton's	202°	49	..	31	20

Composition of Alloys.

Alloy	Aluminium	Antimony	Bismuth	Copper	Iron	Lead	Nickel	Silver	Tin	Zinc	
Admiralty metal				70					1	29	
Ajax metal				81.3		7.3			11		(1)
Aluminium bronze ..	30			70							
Babbitt metal		8		1					91		
Bell metal				81					19		
Brass, English				70						30	
Brass, yellow				65		2				33	
Britannia metal.....		10	2	4					84		
Bronze				78					22		
Bronze coinage				95					4	1	
Constantan				60			40				
Delta metal				56	0.8	0.8				41.6	(2)
German silver				62			15			23	
Gun metal				90					10		
Magnalium	90-98										(3)
Magnolia metal		16.5				83.5					(4)
Manganese bronze ..				75							(5)
Monel metal				26.5	1.5		72				
Muntz metal				66						34	
Nickel coinage				75			25				
Pewter						80			20		
Phosphor bronze				90					9		(6)
Pinchbeck bronze ...				86						14	
Queens metal		8.5	8.5			8.5			74.5		
Regulus of Venus...		34		66							
Solder, aluminium ...						25		30	30	15	
Solder, coarse						66			34		
Solder, common						50			50		
Solder, fine						34			66		
Solder, silver		14		20				66			
Speculum metal				67					32		(7)
Stereotype metal		14				84			2		
Sterling silver				7.5				92.5			
Steam metal				88					10	2	
Tobin bronze				61.3	0.2	0.4			0.9	37.2	
Tombac				86						14	
Type metal		15				77			8		
White metal		12				88					
White metal		11.1		5.6					83.3		

(1) Some phosphorus or arsenic.
(2) 0.8% manganese.
(3) Rest magnesium.
(4) Traces of other metals.
(5) 25% manganese.
(6) 1% phosphorus.
(7) 1% arsenic.

The Metallurgy of Iron and Steel.

The most important, the most abundant, and the cheapest of all known metals is iron. It is practically unknown in nature in the metallic state, but its production from its ores has been known to mankind from prehistoric time. Iron implements were made by the Assyrians, the early Egyptians and the peoples of India and China; and the hardening and tempering of steel appears to have been known to the early Greeks.

The development of the iron industry was very slow up to the middle ages, but each century succeeding the XIVth was marked by distinct advances. The beginning of the modern iron industry dates from the invention of the blast furnace in the early part of the XIVth century. Succeeding this, rapid strides with numerous inventions were made, especially in the purification of pig iron.

Ores.—The principal ores from which iron is obtained are usually more or less impure, and commonly mechanically mixed with mineral matter, such as limestone, clay, quartz, etc. These impurities, collectively known as *gangue,* are in some cases mechanically removed before smelting. The principal characteristics of the most important iron ores are given in the following table:

Characteristics of Common Iron Ores.

Name of Ore	Chemical Formula	Approx. per cent of iron in pure ore	Appearance	Specific Gravity	Hardness
Magnetite	Fe_3O_4	72	Black crystalline mass.	5.5	5.5 to 6.5
Hematite	Fe_2O_3	70	Dark gray to deep red crystalline mass.	5.3	5.5 to 6.5
Limonite	$2Fe_2O_3 3Aq$	60	Light brown to black amorphous mass.	3.6to4.0	5.0 to 5.5
Siderite	$FeCO_3$	48	Gray to yellow-brown crystalline mass.	3.8to4.0	3.5 to 4.5

Magnetite is very magnetic, which property is made use of in removing impurities before smelting. It is the richest iron ore known. Siderite is also known as spathic iron ore.

As mined today these ores contain from 25% to 60% of iron. The principal countries where iron ores are mined are: United States, Germany, Great Britain, Spain, France, Sweden,

Russia, Finland, Austria and Hungary; and of these the American ores are decidedly the richest. The iron ores are widely distributed, occurring in the United States in various localities from coast to coast, the Lake Superior district alone furnishing almost fifty million tons annually, chiefly hematite. Practically none of this ore is smelted locally; it is generally transported long distances, because it can be handled by cheap and rapid labor-saving devices; whereas coke, which is bulky in proportion to its weight, suffers considerable waste in transportation. Some of the Superior ores are soft, earthy, surface deposits and are often worked in open cuts and loaded upon cars by large steam shovels, or sometimes loaded by the "caving method," whereby the ore is allowed to fall by gravity into cars situated below the level worked. The cars on reaching the shore are run out over a long line of ore bins and dumped. The ore vessels which dock alongside these bins are loaded by allowing the ore to flow directly into the hold of the vessel. In a very short time the vessel has received her full cargo and is ready to proceed on her long journey to the blast furnaces. The unloading is accomplished with almost as great speed, by means of mechanical unloading machinery; a cargo of 10,000 tons having been completely discharged in less than five hours. The ore is unloaded in enormous piles, from which it is transported, usually by mechanical means, to storage bins near the blast furnaces.

Production of Iron from the Ore.—The modern method of producing metallic iron from the iron ore consists essentially in deoxidizing the iron, and in separating and removing the mineral matter with which it is associated. This is accomplished simultaneously by heating and smelting the ore in contact with coke and lime in a blast furnace. As the resulting cast iron is carburized by the fuel with which it has been in contact, it is later purified by various processes in the manufacture of steel.

The blast furnace used for making cast iron is an enormous cylindrical shaft 80 to 90 feet high and from 18 to 22 feet wide, lined with silicious fire brick. During operation, the entire length above the hearth is filled with a solid column of lumps of fuel, limestone and ore. This is continually charged through a hopper at the top of the furnace, as the fuel is consumed by the hot blast blown through it and the molten iron and slag collects in the

hearth and is drawn away. The hot blast is forced through the holes or "tuyeres," located directly above the hearth, by immense blowing engines, capable of compressing 50,000 to 65,000 cubic feet of air per minute. This blast enters the furnaces preheated to a temperature of 900° to 1,400° F., and due to the combustion of the coke in the charge, quickly generates a temperature of about 2,700° F., which melts the iron already deoxidized and carburized in its slow descent through the furnace, and fuses the gangue and ash with the lime into a complex molten silicate, known as cinder or slag, both of which trickle down over the rapidly burning coke, into the hearth, the iron at this stage completing its saturation with carbon. Directly below the tuyeres is situated the "cinder notch" or "monkey," through which the molten slag is removed, and at the bottom of the hearth is the iron tap hole for withdrawing the molten iron. To counteract the corrosion of the boshes, or lower part of the furnace, at the very high temperature of the fusion zone, it is cooled by circulating water through water boxes or through pipes imbedded in the lining.

The ascending column of gases, after heating and deoxidizing the ore, still contains about 25% by weight of carbon monoxide and is a very valuable fuel. It is partly used for generating the blast, partly for rolling mill engines, and the rest, about one-third of the total, for heating the blast in hot blast stoves.

The amount of slag produced varies considerably. In smelting the rich Lake Superior ores, it has been as low as one-third, whereas the poorer ores will give as much as twice the iron produced. The chemical composition of *blast furnace slag* varies about as follows:

Silicon dioxide	26 to 35%
Alumina	5 to 15%
Calcium oxide	30 to 50%
Magnesia	5 to 20%
Ferrous oxide	0.3 to 1%
Sulphur	1 to 2%

The molten slag, on account of its lower specific gravity, will always float on top of the iron in the hearth, and is therefore easily removed. This is done by piercing the plug in the cinder notch, when the molten slag flows out and runs down an inclined

iron runner into a large iron ladle, usually on a standard gauge track, so that it can readily be drawn away by a locomotive and poured out into the slag dump. These slags are sometimes used in the manufacture of Portland cement.

As soon as the cinder is removed, the tap hole in the iron notch is opened, and some hundred tons or more of liquid pig iron flows out along with about 30 tons of slag, which is deflected by a skimmer and led into the slag ladle. The heavier pig iron flows under the skimmer and into a number of brick-lined ladles, also on a standard gauge track, and is drawn away by a locomotive to the steel works, or to the pig casting machines to be poured into iron molds, carried on an endless chain. The pig iron chills quickly, being sprayed with water or even submerged in a tank of water, so that it is solid when it reaches the other end of the machine, where it falls into railroad cars.

When the old method of sand casting is used, the molten pig iron flows from long runners in the sand, the so-called sows, into a series of impressions, the pig beds, where it is allowed to cool. After cooling, the iron is broken up into smaller pieces with a sledge hammer and then thrown into railroad cars. The pig iron for the basic open hearth process is cast in metal molds to prevent it from carrying sand into the basic hearths.

Purification of Pig Iron.—Pig iron is too weak and brittle to be used for most engineering purposes, and differs essentially from wrought iron and steel in the large percentage of impurities it contains. It is therefore subjected to processes of purification such as the Bessemer, open hearth, or puddling, which reduce the carbon and eliminate the manganese and silicon. The phosphorus and sulphur are reduced by the basic open hearth and the puddling processes.

In the *Bessemer process,* molten pig iron is converted into steel, by having its carbon, silicon and manganese, and sometimes its phosphorus and sulphur, oxidized and then removed. The Bessemer converter is a huge retort lined with silicious material, clay or dolomite, and of a capacity usually ranging up to 20 tons. It is hung and turned on trunnions, through one of which the blast is carried to the tuyeres at the bottom. Immediately after the requisite charge of molten pig iron has been run in, cold air is forced through the molten mass at the rate of 20,000 to 30,000

cubic feet per minute. Due to the oxidation of the silicon* and other constituents contained in the charge, the initial temperature of the molten mass, which was about 2,000° F., is rapidly raised to a point well above that of the melting point of the resulting steel, or about 2,700° F. In a few minutes, the manganese and silicon are oxidized and have formed a slag. The carbon begins to oxidize, forming carbon monoxide, which boils up through the molten metal and escapes at the mouth of the converter, where it burns with a brilliant flame. In from 10 to 20 minutes, the flame falls, the carbon having been reduced to the lowest practical limit, and the operation is stopped. Though it is possible to judge the end of the process from the appearance of the flame, and thus manufacture steel of any desired carbon content from 0.10 to 1.25%, it is almost universal to continue the blow until the decarburizing is complete, and then add enough carbon to give the desired content. In making low carbon steel, recarburizing is not needed, but in both instances considerable quantities of manganese are added to remove the absorbed oxygen from the molten steel. For low carbon steel the manganese is added as ferro-manganese, whereas when the carbon content is to be increased, spiegeleisen is added. In both instances the manganese is oxidized and forms a slag. The molten steel is then poured into a large clay-lined casting ladle, through the bottom of which the steel is run into ingot molds.

In the *acid open hearth process,* steel is made by the melting together and purifying of pig iron and scrap iron in a shallow basin-like reverberatory furnace lined with a silicious material. It is heated by burning within it gas and air which has been strongly preheated. This process is naturally much slower, as the transference of heat from the flame to the charge, through the layer of slag is necessarily slow. The impurities are oxidized by the excess of oxygen in the flame playing on the surface of the molten metal, and, if equal parts of pig iron and scrap steel are used, this requires about 8½ hours in the acid process and 11½ hours in the basic. The process may be hastened by adding light charges of ore (ferric oxide), the oxygen of which combines with

* In the basic Bessemer process, phosphorus and silicon are the principal sources of heat. The heat generated by the oxidation of the silicon is consumed in heating the lime needed to absorb the phosphorus and sulphur.

the carbon. In some foreign furnaces little or no scrap steel is used, and the time required for the purification is prolonged to 17 hours.

The *basic open hearth* is lined with a basic material, usually calcined dolomite, and the process is similar to the acid open hearth with the exception that lime is added to form a very basic slag. This slag combines with the oxidized phosphorus and sulphur, thus removing these, for some steels, very undesirable elements.

In the *puddling process,* the pig iron is remelted in small charges on the shallow hearth of a reverberatory furnace lined with iron ore. During the heating, iron oxide is frequently stirred into the molten metal by the workman or "puddler." This facilitates the oxidation of the silicon and the sulphur, both of which combine with the iron oxide to form a very basic slag, known as "puddling cinder" or "tap cinder." The carbon, which is oxidized to carbon monoxide, escapes and burns in small jets over the molten mass. As the metal is gradually purified, its melting point slowly increases to a point above that of the temperature of the furnace, with the result that the molten metal assumes a pasty granulated condition, and "comes to nature," as it is called. The puddler then works the mass with his hook or rabbler into balls weighing from 100 to 200 pounds. These balls of semi-solid slag-bearing iron are removed and worked with a steam hammer or in revolving squeezers which rotate the puddled ball so that turning around on its own axis it is changed into a rough cylindrical mass, still white hot. It is then immediately rolled down to flat bars, called "muck bars," and cut into short uniform lengths and piled up into a bundle, the layers alternately laid cross-wise and finally bound together with wire. These bundles are then raised to a welding heat and again rolled out into bars, known as merchant bars. This further rolling has the effect of removing still more slag and of producing in the iron a cross-wise network of slag-covered fibres. The manufacture of wrought iron is still done by hand and requires great strength and endurance on the part of the puddler who stirs and balls the pasty mass. The small charges are turned out very slowly, and this combined with the high wages commanded by the puddler in a measure explains the high market price of the raw product.

Wrought iron is converted into steel by adding carbon to it while it is in the solid state or after it has been melted. These are known as the cementation process and the crucible process, respectively.

In the *cementation process,* carbon is allowed to soak into the iron bars at a bright-red heat. The process is very slow, the rate of absorption being approximately ⅜ inch in 24 hours. The furnace used is of a very low form, built of stone and surmounted by a superstructure or chimney to carry off the products of combustion, and to reduce the loss of heat by radiation. The converting pots, of which there are two in each furnace, resemble oblong boxes made of stone and are from 2½ to 4 feet wide and deep and from 8 to 15 feet long. The bottoms of the boxes are first covered with a layer of charcoal and on the top of this are placed the wrought iron bars, which in turn are covered with charcoal, then a layer of bars covered with charcoal and so on until the pots are filled; this requires from 10 to 30 tons, depending upon the size. The pots are then heated and the fire started. In about two days a temperature of 1,200° to 1,800° F. is reached, and this is maintained for a period of up to 11 days, depending upon the grade of steel made. The following grades are made at Sheffield:

1. Spring heat containing 0.50% carbon.
2. Country heat containing 0.60% carbon.
3. Single-shear heat containing 0.75% carbon.
4. Double-shear heat containing 1.00% carbon.
5. Steel-through heat containing 1.25% carbon.
6. Melting heat containing 1.50% carbon.

When the cementation has reached the desired point, the fire is stopped and the furnace allowed to cool, this requiring about a week, when the converting pots are opened and the bars taken out. The surface of the bars has now become covered with small blisters, due to the formation of gas in the wrought iron during the heating, which has given it the name of "blister steel." For the manufacture of finest quality of tool steel, the blister steel is remelted by the crucible process, which completely separates the slag from it.

The *crucible process,* which has now to a great extent superceded the cementation process, consists in melting the wrought

iron or blister steel in covered clay or graphite crucibles, which have a capacity of from 50 to 100 pounds, the required amount of carbon being placed on top of the charge before melting. The melting takes place either in coke furnaces, each holding two crucibles which are surrounded by a coke fire, or in gas furnaces heated by the fire from a mixture of gas and air which is preheated in chambers before it enters the melting hole. The melting in either case requires from 2 to 3 hours, but the crucible is left in the furnace about ½ hour longer until it is "dead," i. e., can be poured without giving off gas. This is necessary in order to obtain solid ingots. If the "killing" time has been too long, the resulting steel will be weak and brittle, probably due to the absorption of too much silicon. At the proper time the crucibles are removed from the furnace, the slag is swabbed off and the steel poured into ingots.

Of late, *electric steel making processes* have been introduced, and the product appears to be generally better though much more costly than that produced by both the Bessemer and the open hearth processes. There are two different forms of furnaces used, those heated by electric arcs, of which the Heroult furnace is the best known, and those of the induction type where the metal is heated by its own resistance to the electric current; of the latter, the Roechling-Rodenhausen and the Kjellin furnaces are good examples. In all the different types the electric current has nothing to do with the purification, it being only a source of heat. The main advantage which these furnaces possess over the Bessemer and the open hearth processes is that a more complete removal of sulphur, oxide of iron and suspended slag is obtained. The latter is due to the very tranquil condition of the molten mass, which permits the finely divided slag to rise to the surface.

Properties of Steel.

The various properties of steel are considerably influenced by the quantity of certain substances, namely, carbon, silicon, manganese, phosphorus and sulphur, occurring as impurities in the pig iron and either partly or almost completely removed during the various processes of purification (see page 135).

Tempering.—Steel is subject to considerable modification

by a thermal treatment or tempering. Thus when it is heated to redness and suddenly chilled it becomes extremely hard, whereas when it is slowly cooled it becomes comparatively soft. The hardness of steel may be reduced to any desired degree by a subsequent careful heating followed by a sudden chilling. The temperature at which this change occurs can be roughly judged by observing the interference colors produced on polished steel by the thin film of oxide which forms on the surface. The following examples, based on the figures obtained by Claudel, Rose and others will serve to illustrate this:

Temper Colors for Various Tools, Etc.

Pale yellow	— 430°—440° F. —	Hammer faces. Planer tools for iron and steel. Razor steel.
Distinct yellow	— 450°—460° F. —	Screw cutting dies. Taps. Drills. Punches.
Brown	— 480°—500° F. —	Brass drills. Shears.
Purple	— 525°—550° F. —	Augers. Surgical instruments. Cold chisels. Table knives.
Blue	— 550°—575° F. —	Gimlets. Springs. Needles. Axes.
Black Blue	— 575°- 600° F. —	Saws for wood.

Case-hardening is a process which produces a thin and extremely hard outer surface surrounding a softer and more malleable interior, and consists in superficially carburizing the steel in contact with charcoal, sugar, potassium cyanide, etc. Various hydrocarbons, among others, fuel gas, in specially constructed furnaces have also been used. The carbon begins to penetrate the steel at about 1,300° F. and reaches a maximum under 1,600° F. The process usually lasts from 5 to 48 hours. After carburizing, the steel is annealed, then reheated and suddenly chilled.

Alloy Steels.

When the properties of a steel are due chiefly to the addition of one or more elements other than carbon it is known as an alloy steel. Of these, two different kinds, the ternary and the quarternary alloy steels, have been produced. The *ternary alloy steels* are those which are composed of iron and steel and one other element affecting its properties. The most important of these are: nickel-steel, manganese-steel, chrome-steel, vanadium-steel, titanium-steel, tungsten-steel and aluminium steel. The *quarternary alloy steels* contain two elements of influence in addition to iron and carbon. Examples of these are: nickel-manganese steel, nickel-chrome steel, nickel-vanadium steel, nickel-tungsten steel, tungsten-manganese steel, chrome-vanadium steel and tungsten-chromium steel. They are usually manufactured by adding the alloying element at the end of the purification process in much the same manner as carbon is added during recarburizing. A few of the alloy steels are briefly described below.

Nickel-steel is the most common alloy steel and is used for steel bridges, rails, steel castings, engine forgings and engine shaftings, wire cables, automobile parts, etc. It possesses a much higher elastic limit and a slightly lower ductility than ordinary carbon steels. In addition, it has a much greater resistance toward corrosion, a higher compression strength and a greater resistance to impact. It usually contains from 3.0 to 3.5% of nickel and from 0.2 to 0.5% carbon.

Invar is a patented nickel-steel alloy which contains about 36% of nickel, about 0.4% manganese and about 0.2% carbon. Its coefficient of expansion is the lowest of all known metals and alloys, varying, according to the Bureau of Standards in Washington, from 0.000000208 to 0.000000244 for 1° F. Invar is used mostly for temperature-sensitive scientific apparatus, chronometers, clock pendulums and for survey tape measures, which can be used for all except the most accurate survey work without a correction for temperature.

Platinite is a patented nickel-steel alloy containing about 42% of nickel, which has practically the same coefficient of expansion and contraction at atmospheric temperatures as glass and platinum. Platinite is used for the wire network enclosed in "ar-

mored" glass, and in place of platinum in the electric connections which pass through glass in the base of incandescent lamps.

Manganese-steel is used chiefly for railroad rails and frogs, burglar-proof safes, the wearing parts of rock crushing machinery, links on chain elevators, automatic car couplers, horse-shoes, etc. It contains on an average 12 to 15% of manganese, but shows considerable variation in carbon, from less than 0.5 to 2.0%, though 0.5% seems to be the most desirable. Manganese-steel welds with great difficulty, and its castings are remarkably free from blow-holes. In opposition to carbon-steel, it is almost as brittle as glass immediately after casting, and becomes as ductile as wrought iron, when rapidly chilled from a temperature of 1,800 to 1,900° F. Its hardness cannot be decreased by annealing. It has a high electrical resistance, which varies but slightly with changes in temperature, and a low thermal conductivity.

Chrome-steel finds comparatively little use outside of the manufacture of armor plate and armor-piercing projectiles. It has, to some extent, been used for automobile steels especially by French makers, but very little, if any, finds its way into American cars. Chrome steel usually contains about 2% of chromium and from 0.8 to 2% of carbon. It is characterized by an intense hardness, a high elastic limit and a considerable resistance against impact.

Tungsten-steel or **Mushet-steel** is used chiefly as a tool steel and for the manufacture of steel magnets, as it retains magnetism longer than ordinary steel. It usually contains from 5 to 10% of tungsten and from 1 to 2% of carbon. It is considerably harder and less brittle than ordinary carbon steels.

Chrome-Tungsten steel is characterized by the very valuable property of red hardness, i. e., retaining its hardness when heated to a dull redness (about 1,100° F.). It is used chiefly as a high-speed tool steel. Its composition varies considerably but is considered best as follows: Tungsten about 18%, chromium about 6%, manganese about 0.1%, and carbon about 0.7%.

Vanadium-steel has a very high tensile strength, and is especially useful for automobile parts, engine and machine parts, gun barrels, etc. Its composition varies, but is generally

Relative Strength and Fatigue Resistance of Steels.*

No.	Material.	Heat Treatment.	Approximate Analysis. C.	Mn.	Cr.	Ni	V.	Elastic Limit. (E.)	Tensile Str'gth	Elong. in 2 in. %.	Red. of Area, %. (R.)	Alternations, (A.)	Quality E×R×A + 10^6
			Elastic Limits, 30,000 to 60,000.										
1	Wrought iron	R.	0.05	0.05				32,020	49,450	42.5	53.8	583	1004
2	Mild Va steel	A.	0.11	0.31			0.19	52,120	54,400	44.0	60.1	1111	2143
3	Old plate	R.	0.24	0.42				58,160	70,840	29.5	53.4	612	1901
4	Mild steel	A.	0.18	0.40				39,460	60,650	35.0	62.6	871	2152
5	Mild steel	O. T.	0.18	0.40				45,390	70,050	33.0	64.0	777	2258
6	Forging steel	A.	0.26	0.28				43,010	61,850	34.0	63.4	1415	3858
7	Forging steel	O. T.	0.26	0.28				52,250	77,310	28.0	65.3	1175	4008
8	Va case hardening	A.	0.15	0.25			0.12	44,790	55,990	49.0	69.0	1958	6051
9	Cr-Ni steel	A.	0.36	0.34	0.90	1.70		56,520	81,370	32.0	68.5	978	3787
10	Steel casting	A.	0.18	0.65				34,690	58,800	28.0	44.9	269	419
11	Va steel casting	A.	0.19	0.60			0.076	44,340	70,250	29.5	44.9	850	1671
12	Cr-Va casting	A.	0.57	0.68	0.75		0.16	52,330	92,900	16.0	20.5	656	704
			Elastic Limit, 60,000 to 100,000.										
13	Ni forging steel	A. T.	0.21	0.45		3.70		61,140	79,700	30.0	62.5	746	2851
14	Ni forging steel	O. T.	0.21	0.45		3.70		77,140	98,470	25.0	63.3	664	3545
15	Cr-Va forging	A.	0.26	0.50	1.00		0.16	61,920	92,900	29.0	57.3	1606	5706
16	Spring steel	A.	1.00	0.30				63,800	125,000	8.5	15.2	1260	1222
17	Cr-Va spring	A.	0.40	0.77	1.22		0.19	67,520	100,600	26.0	61.7	1406	5858
18	Cr-Ni-Va steel	A.	0.30	0.27	1.51	3.45	0.085	69,140	96,880	28.5	58.5	507	2402
19	Ni-Va steel	A.	0.24	0.72		3.40	0.15	79,260	99,700	25.0	54.0	798	4046
20	Cr-Ni-Va steel	A.	0.57	0.27	0.95	2.04	0.07	95,150	129,100	21.0	49.8	983	4659
21	Cr-Ni steel	T.	0.37	0.34	0.89	1.70		86,000	102,700	22.0	63.8	702	3695
			Elastic Limits, 100,000 and Over.										
22	Spring steel	O. T.	1.00	0.30				101,000	186,300	9.5	16.1	561	912
23	Cr-Va spring	O. T.	0.40	0.77	1.22		0.19	195,300	208,500	10.0	36.3	480	3403
24	Cr-Va forging	O. T.	0.30	0.50	1.00		0.16	141,600	151,750	16.0	56.2	717	5705
25	Ni-Va steel	O. T.	0.24	0.72		3.40	0.15	129,900	134,600	18.0	54.8	626	5270
26	Cr-Ni-Va steel	O. T.	0.30	0.27	1.51	3.45	0.085	152,300	159,900	17.0	56.9	467	4369
27	Cr-Va spring	O. T.	0.40	0.77	1.22		0.19	183,400	187,600	14.0	50.6	634	5883
28	Cr-Ni steel	T.	0.36	0.34	0.95	1.70		134,500	150,300	15.5	53.5	579	4166

NOTES. — Static tests made on samples ½ in. diam. × 2 in. long. Dynamic tests made on the Landgraf-Turner alternating impact machine (old form). The quality figure is the product of: Elastic limit, representing useful strength; reduction of area, representing static ductility; dynamic figure, representing fatigue-resisting property — divided by 1,000,000 = E × R × A + 10^6.

Heat treatment, R., raw; A., annealed; O. T., oil tempered; T, tempered. Nos. 9, 17, 18, 20, 21, 26, 27, 28 were crucible steels, the others open-hearth.

* Kent "Mechanical Engineers' Pocket Book," article by W. L. Turner, "Iron Age," July 2, 1908.

considered to give best results when it contains from 0.10% to 0.20% of vanadium. With the exception of carbon, vanadium has the most powerful effect on steel. The vanadium steels weld quite readily and are forged without difficulty, though they must be heated slowly and worked carefully at first. Vanadium is especially effective when added to nickel and chromium steel, as shown by the table on page 134.

The Effect of Various Impurities Contained in Pig Iron.

Combined carbon when present up to 1% increases hardness, brittleness and shrinkage.

Graphitic carbon produces soft and weak castings when over 3%.

Silicon produces softness and counteracts the hardening effect of sulphur. It increases fluidity and reduces shrinkage. Above 3.5%, the iron becomes brittle and weak. These effects are modified by sulphur and manganese.

Sulphur increases hardness, brittleness and shrinkage. Iron containing more than 0.1% is very weak and brittle.

Manganese counteracts sulphur when less than 0.5% is present. Above 2% it hardens the iron decidedly. Iron and manganese mix in almost all proportions.

The alloy spiegeleisen contains from 10 to 30%, and ferro-manganese contains over 50% of manganese. Manganese increases the solubility of carbon in iron, pig iron usually containing 3.5 to 4.25% carbon, rarely over, spiegeleisen as high as 5% and ferro-manganese 6%. Manganese increases shrinkage.

Phosphorus increases fluidity and softness, and decreases shrinkage. Above 1% it decreases strength.

Aluminium increases the softness and strength of white iron and softens and weakens gray iron.

Copper is without any appreciable effect on pig iron when present below 1%.

Titanium when present from 0.2 to 0.3% increases the strength about 25%. It acts as a deoxidizer and purifier, removing nitrogen, and does not remain in the iron.

Vanadium acts as a deoxidizer and alloys with the iron. The presence of 0.15% considerably increases the strength.

Combustion

Combustion, as commonly understood, is the rapid chemical union of a substance with the oxygen of the air, accompanied by the generation of light and heat. Such a substance is called a *combustible,* and if used for heating purposes, it is called a *fuel.*

CHEMISTRY OF COMBUSTION.

Principal Elements of Combustion.

The elements which play a more or less active part during combustion, either as constituents of the combustible or of the air, are carbon, hydrogen, oxygen, nitrogen and sulphur, the two elements first mentioned being the most important constituents of fuels.

Carbon is a solid, and is the principal constituent of coal. It exists in a perfectly pure state in graphite and diamond. In chemical combination with hydrogen, carbon forms the so-called hydrocarbons, which are found in nature as gases, oils and tars. Among these are natural gas and petroleum. In coal, the hydrocarbons form the so-called volatile matter while the carbon which is not united with hydrogen is called fixed carbon. Combined with oxygen and hydrogen and sometimes a small quantity of nitrogen, carbon forms the vegetable substances, such as wood and peat.

Hydrogen, unlike carbon, is not found in nature in its elementary form, but is always combined with other elements. It is combined with oxygen to form water, and with carbon, forming the numerous hydrocarbons. In the different kinds of coal, carbon and hydrogen exist either combined, or they combine when the coal is brought to the point of ignition, and combustible hydrocarbons are produced.

Oxygen and **nitrogen** are mainly important as the constituents of air. In fuels they are present only in combination with other elements.

Sulphur is present in most coals, but generally in very small quantities only. In coal it is usually present in the form of iron pyrites, but occasionally also as calcium sulphate. The presence of sulphur in fuel is not desirable, since it favors the formation of clinkers, and its product of combustion may attack iron. Sulphur is a combustible, but its heating value is slight. When present as calcium sulphate, it has no heating value.

Perfect and Imperfect Combustion.

When a fuel is burned, the products of combustion as well as the heat generated, are not always the same. Combustion may be perfect or imperfect, and this depends, with other factors that are mentioned later, mainly on the supply of air and the temperature in the furnace.

When a sufficient supply of air, and consequently of oxygen, of a high temperature is used, the carbon of the fuel unites with the oxygen to form carbon dioxide CO_2 (also called carbonic acid gas), each pound generating 14,600 B. T. U., while the hydrogen burns to steam, generating 62,000 B. T. U. per pound of hydrogen. When the maximum amount of heat is produced, combustion is complete.

If too little air is used, the quantity of oxygen is not sufficient to produce carbon dioxide from the carbon, but another combination is formed, called carbon monoxide, CO. Considerably less heat is generated by this incomplete combustion, only 4,450 B. T. U. being produced. It follows, that by incomplete combustion of carbon more than 10,000 B. T. U. are lost per pound, or about 70%.

Properly speaking, there is no incomplete combustion of hydrogen, but it may happen that some of the hydrocarbons containing the hydrogen, instead of being ignited, are simply distilled from the burning coal and escape through the chimney. Sometimes they are partly burned, and the carbon they contain is separated and either deposited as soot, or, being in a very finely divided condition, is carried away by the hot combustion

gases and appears as smoke. Though not burned, this carbon is often heated to such a temperature that it begins to glow. It then renders the flame luminous, and this is always a sign that combustion is not complete.

If under such conditions more air is supplied, the highly heated carbon burns completely to carbon dioxide, the color of the flame disappears and combustion is perfect. If on the other hand the luminous flame comes in contact with some cold surface, its temperature may be reduced so that the carbon ceases to glow and is deposited as soot.

Heating Value of One Pound of Various Substances.

	B. T. U.
Acetylene (C_2H_2) to CO_2 and H_2O	21,500
Benzole gas (C_6H_6) to CO_2 and H_2O	18,000
Carbon to carbon monoxide, CO	4,450
Carbon (wood charcoal) to carbon dioxide, CO_2	14,600
Hydrogen to water, H_2O	62,000
Marsh gas (methane CH_4) to carbon dioxide and water, CO_2 and H_2O	23,600
Olefiant gas (ethylene C_2H_4) to CO_2 and H_2O	21,500
Sulphur to sulphur dioxide, SO_2	4,050

Volume of One Pound and Specific Heat of Various Gases.

62° F. and 30 inches of Mercury.

	Volume in Cubic Feet.	Specific Heat at Constant Pressure.
Acetylene	14.6	.....
Air (atmospheric)	13.16	0.2375
Carbon dioxide	8.627	0.2169
Carbon monoxide	13.55	0.2426
Hydrogen	189.8	3.409
Marsh gas	23.72	0.593
Nitrogen	13.55	0.2438
Olefiant gas	13.55	0.404
Oxygen	11.86	0.2175
Sulphur dioxide	5.93	0.1544

Various figures for the above two tables are given by different authors. The first table represents a fair average, and the figures are sufficiently close for practical purposes. In the second table, the figures on volume are from the Smithsonian Tables, whereas those on specific heat are from Regnault.

Theoretical Amount of Air Necessary for Complete Combustion.

Air is a mixture of oxygen and nitrogen. One hundred cubic feet of air consist of approximately 21 cubic feet of oxygen and 79 cubic feet of nitrogen. One hundred pounds of air consist of approximately 23 pounds of oxygen and 77 pounds of nitrogen. There are other gases present in the atmosphere in small quantities. For detailed composition of air, see chapter "Chemistry," page 99.

From these figures it follows that every cubic foot of oxygen consumed in combustion requires 4.76 cubic feet of air, and that every pound of oxygen corresponds to 4.35 pounds of air.

Air at 62° F. and 30 Inches Pressure Required for Complete Combustion of One Pound of Each of Combustible Elements of Fuel, and Products of Combustion.

Element.	Air.		Products.
Hydrogen..	34.8 lbs.	457 cu. ft.	9.0 lbs. of water 26.8 lbs. of nitrogen 35.8 lbs.
Carbon ...	11.6 lbs.	152 cu. ft.	3.66 lbs. of carbon dioxide 8.94 lbs. of nitrogen 12.6 lbs.
Sulphur ...	4.35 lbs.	57 cu. ft.	2.0 lbs. of sulphur dioxide 3.35 lbs. of nitrogen 5.35 lbs.

Calculation of Theoretical Amount of Air.

The theoretical quantity of air in pounds that is consumed in the complete combustion of one pound of fuel of known composition, is found as follows:

$$\text{Pounds of air} = \frac{C \times 11.6 + (H - \frac{1}{8} \times O) \times 34.8}{100} \text{ or}$$

$$\text{Pounds of air} = [C + 3 \times (H - 0.125 \times O)] \times 0.116$$

C = per cent of carbon by weight.

H = per cent of hydrogen by weight.

O = per cent of oxygen by weight.

If the amount of air is to be calculated in cubic feet, the following formulas may be used:

$$\text{Cubic feet of air} = \frac{C \times 152 + (H - \frac{1}{8} \times O) \times 457}{100} \text{ or}$$

$$\text{Cubic feet of air} = [C + 3 \times (H - 0.125 \times O)] \times 1.52$$

Not all of the hydrogen in a fuel is available for combustion, since part of it is combined with the oxygen contained in the fuel. When calculating the amount of air necessary for combustion, or as will be shown later, the heat developed, this must be taken into consideration, i. e. the combined hydrogen must be subtracted from the total hydrogen present. As oxygen combines with $\frac{1}{8}$ of its weight of hydrogen, the quantity of hydrogen available for combustion is found by subtracting from the total hydrogen $\frac{1}{8}$ the weight of the oxygen, i. e., $H - \frac{1}{8} \times O$.

Example.—A coal contains 80% of carbon, 4½% of hydrogen and 4% of oxygen. How many pounds of air are theoretically necessary for complete combustion?

Solution.—$[80 + 3 \times (4.5 - 0.125 \times 4)] \times 0.116 =$ 10.67 pounds of air, or $[80 + 3 \times (4.5 - 0.125 \times 4)] \times 1.52 =$ 139.8 cubic feet of air.

Quantity of Gaseous Products of Combustion.

The total quantity of the gaseous products from the complete combustion of one pound of fuel is calculated as follows:

Pounds of gas = $0.126\,C + 0.358\,H$.

Cubic feet of gas = $1.52\,C + 5.52\,H$.

Example.—What quantity of gaseous products in weight, and in volume at 62° F., is formed by the complete combustion of a

pound of coal which contains 80% of carbon and 4% of hydrogen?

Solution.—Weight of gas $= 0.126 \times 80 + 0.358 \times 4 =$ 11.512 pounds.

Volume of gas $= 1.52 \times 80 + 5.52 \times 4 = 143.68$ cubic feet.

To find the volume at other temperatures than 62°F., the following formula is used:

$$V_1 = V \times \frac{t_1 + 460}{522}$$

V_1= volume at desired temperature.

V = volume at 62° F.

t_1 = desired temperature.

Example.—The volume of the combustion gases at 62°F. is 143.68 cubic feet. What is the volume at 450° F?

Solution.—Volume at 450° F. $= 143.68 \times \frac{450 + 460}{522} =$ 250.48 cubic feet.

Air Necessary in Actual Practice.

For good combustion in boiler furnaces, 60% to 100% more air is required than the theoretical amount mentioned above. The best results have been obtained with 19 pounds of air per pound of Pocahontas coal. The average consumption of air, however, can safely be taken as 100% more than the theoretical, and in a great number of cases this amount of air is even exceeded, which of course is a great loss, as all the excess air which passes through the fire must be heated from the boiler-room temperature to that of the escaping gases.

If the per cent of carbon dioxide in the escaping gases is known, the amount of *excess air* can be calculated, using Bunte's formula,

$$R = \frac{18.9}{P}$$

R=ratio of actual air to theoretical amount required.

P=per cent of carbon dioxide in gases.

18.9=per cent of carbon dioxide in gases, in theoretically complete combustion.

Combustion may be considered practically complete, however, if the amount of CO_2 in the chimney gases amounts to 15%. The

average per cent of carbon dioxide found in chimney gases varies from 6% to 15%, and 6% is not a very rare occurrence. Applying Bunte's formula, it indicates that $\frac{18.9}{6}$ or 3.15 times as much air was used as was theoretically necessary.

The *waste of heat* caused by using 3.15 times the theoretical quantity of air is considerable. If we assume the air to enter the furnace at 60°F. and to leave the boiler at 500°F., and if we assume the specific heat of the escaping gases to be 0.24, then the amount of heat carried away by this air, would be $(500 - 60) \times 0.24 \times 3.15 \times 12.6 = 4{,}191.3$ B. T. U. per pound of coal. If we consider that a pound of coal can furnish 8,400 B. T. U. with the proper amount of air, then the percentage of waste amounts to about 50%. From this it can readily be seen, that in boiler furnaces a greater waste is caused by an excess of air then by incomplete combustion. Engineers and firemen should therefore try to burn their fuel with as little air as will prove sufficient.

Conditions for Complete Combustion of Fuel.

The important conditions necessary to bring about the complete combustion of fuel are:

1. A proper amount of air.
2. Thorough mixture of combustible gases and air.
3. Bringing together air and combustible gases at the highest possible temperature.

Calculation of Heating Value of Fuels.

The heating value of fuels can be calculated by means of Dulong's formula, employing approximate figures:

Heating value in B. T. U. =

$$\frac{14{,}500^{*} \times C + 62{,}000 \times (H - \frac{1}{8} \times O) + 4{,}050 \times S}{100}$$

C = Per cent of carbon in fuel.
H = Per cent of hydrogen in fuel.
O = Per cent of oxygen in fuel.
S = Per cent of sulphur in fuel.

* Some authors use 14,600.

Heating Values of Various Fuels.*

Fuel.	Theoretical Heating value in B. T. U.	Theoretical Evaporative power, in pounds of water from and at 212° F.
Anthracite coal	14,600 to 14,800 per pound..	15 to 15.3 per pound
Semi-anthracite coal.	14,700 to 15,500 per pound..	15 to 16 per pound
Semi-bituminous coal	15,500 to 16,000 per pound..	16 to 16.5 per pound
Bituminous coal.....	13,500 to 15,500 per pound..	14.5 to 16 per pound
Lignite	11,000 to 13,500 per pound..	11.3 to 14.5 per pound
Coke	12,400 to 13,400 per pound..	12.8 to 13.8 per pound
Wood, kiln-dried	7,200 to 8,200 per pound..	7.5 to 8.5 per pound
Wood, air-dried (25% water)	5,100 to 5,900 per pound..	5.25 to 6 per pound
Petroleum	18,700 to 20,900 per pound..	19.25 to 21.5 per pound
Natural gas	890,000 to 1,100,000 per 1,000 cub. ft.	915 to 1,130 per 1,000 cub. ft.
Coal gas.....	660,000 to 735,000 per 1,000 cub. ft.....	680 to 755 per 1,000 cub. ft.
Water gas.....	280,000 to 320,000 per 1,000 cub. ft.....	288 to 330 per 1,000 cub. ft.
Producer gas.....	140,000 to 160,000 per 1,000 cub. ft.....	144 to 165 per 1,000 cub. ft.

A simpler formula which gives very satisfactory results is the following:

Heating value in B. T. U. $= (C + 4.28 \times H) \times 145$.

This means that the percentage of hydrogen is multiplied by 4.28, added to the percentage of carbon, and the sum multiplied by 145.

Example.—If a coal contains 80% of carbon, 4.5% of hydrogen and 4% of oxygen, what is its heating value?

Solution.—According to Dulong's formula the heating value is:

$$\frac{14{,}500 \times 80 + 62{,}000 \times (4.5 - 0.5) + 4{,}050 \times 0}{100} = 14{,}080 \text{ B.T.U.}$$

Using the second short formula, the following result is obtained:

* The values for the coals are from Kent "Mechanical Engineers' Pocket Book."

Heating Value $= (80 + 4.28 \times 4.5) \times 145 = 99.26 \times 145 =$ 14,393 B. T. U.

The theoretical evaporative power of a pound of fuel from water of 212°F. into steam of 212°F. can be found by dividing the heating value of the fuel by 970.4, the latent heat of evaporation at 212°F.:

Evaporative power in pounds of water $=$

$$\frac{(C + 4.28 \times H) \times 145}{970.4}$$

Since $\frac{145}{970.4}$ = approximately 0.15, the formula may be simplified as follows:

Evaporative power in pounds of water $= (C + 4.28 \times H) \times 0.15$.

Example.—What is the theoretical evaporative power of a coal, containing 80% of carbon and 4.5% of hydrogen?

Solution.—$(80 + 4.28 \times 4.5) \times 0.15 = 99.26 \times 0.15 = 14.89$ pounds of water per pound of coal.

Temperature of the Fire.

The ideal temperature of the fire can be calculated by dividing the number of B. T. U. in one pound of the fuel by the pounds of combustion gases multiplied by the specific heat. For instance, if we assume that a pound of coal contains 14,500 B. T. U. and is burned with 19 pounds of air, forming 20 pounds of gas, and also assume the mean specific heat of the gases to be 0.24, then the ideal temperature of the fire is $\frac{14{,}500}{20 \times 0.24} = 3{,}020°$F. By actual measurement, however, it has been found to be some-

Heating Degrees.

	Cent.	Fahr.
Red .....commences at	525°	955°
Dark red .....commences at	700°	1292°
Cherry red .....commences at	850°	1562°
Light red .....commences at	900°	1652°
Yellow .....commences at	1100°	2012°
White .....commences at	1300°	2372°
Full white .....commences at	1500°	2732°

what lower. With a stationary boiler being under-fired with coal, and having a large grate surface of 26 square feet, the temperatures in degrees F. were found to be:

Over the center of the fire at different times	3200°	3610°	3405°
Over the bridge	1730°	1730°	1735°

FUELS.

Fuels are divided into three main classes, solid, liquid and gaseous. Some fuels used for the production of steam are coal, coke, wood, charcoal, peat, sawdust, refuse tan bark, straw, bagasse or refuse sugar cane, asphalt, creosote, petroleum, oil, alcohol, artificial gas and natural gas.

Coal.

The following classification of coals is a good one:*

	Fixed Carbon %	Volatile Matter %
Anthracite	97 — 92.5	3 — 7.5
Semi-anthracite	92.5 — 87.5	7.5 — 12.5
Semi-bituminous	87.5 — 75	12.5 — 25
Bituminous	75 — 50	25 — 50
Lignite	Under 50	Over 50

Bituminous coal may be further divided into caking, non-caking and cannel coal.

Anthracite Coal.—This coal contains a very high percentage of carbon. The specific gravity varies from 1.35 to 1.92. It retains its form when exposed to the temperature of ignition, but when heated too rapidly falls to pieces. The flame is generally short, and of a bluish yellow color. The coal is ignited with difficulty, yields an intense local or concentrated heat, and combustion becomes extinct, while a considerable quantity of the fuel still remains on the grate.

Semi-anthracite and **semi-bituminous** are coals between bituminous and anthracite, deriving their name from whichever class they are nearest to.

Dry Bituminous Coal.—This is a freely burning coal, and is generally lighter than anthracite. It contains a relatively small proportion of volatilizable matter, about 15 per cent, and

* Kent "Mechanical Engineers' Pocket-Book."

quickly arrives at the temperature of ignition. It swells considerably in coking, thus facilitating the access of air and the rapid and complete combustion of the fixed carbon. In some cases where combustion is slow, the masses of coke scarcely cohere, and the original forms of the pieces of the coal are in some measure preserved.

Bituminous Caking Coal.—This coal has the same range of specific gravity as the dry bituminous coal. It contains the maximum proportion of volatilizable matter, averaging about 30 per cent of the whole weight. It develops much of the hydrocarbon gases, and burns with a long flame. It swells considerably and gives a coherent coke, which preserves nothing of the original form of the coal.

Long-Flaming or Cannel Coal.—This coal ignites readily and burns with a bright flame. It is valuable as an enricher in gas-making, on account of its high percentage of hydrocarbons. Some of these coals are caking, others are non-caking.

Lignite.—Brown lignite is sometimes of woody texture, sometimes earthy. Black lignite is either of woody texture, or it is homogeneous, with a resinous fracture. The coke produced from various lignites is either pulverulent like that of anthracite, or it retains the form of the original fibers. Lignite is less dense than coal. It yields about 47 per cent of coke.

The various kinds of coal are distributed over the United States as follows:

Anthracite		Eastern part of Allegheny Mountains and Rocky Mountains in Colorado.
Bituminous Coal	Caking	Mississippi Valley.
	Non-Caking	Maryland and Virginia.
	Cannel	Pennsylvania, Indiana and Missouri.
Lignite		Colorado, Kentucky and Washington.

The following table gives the composition of some American coals*:

* Sturtevant "Mechanical Draft"; and Technical Publishing Co. "Practical Engineer."

Composition of Various American Coals.

Name or Locality.	Constituents in per cent of Total Weight.				
	Moisture.	Volatile Matter.	Fixed Carbon.	Ash.	Sulphur.
Arkansas—					
Coal Hill, Johnson Co.	1.50	12.17	74.50	8.79	3.04
Huntington Co.	1.30	17.75	72.35	7.90	0.70
Lignite	32.00	28.50	34.50	5.00	
Jenny Lind, Sebastian Co.	1.26	17.64	72.48	8.62	2.11
Spadra, Johnson Co.	1.47	13.27	78.63	6.63	1.60
Colorado—					
Bituminous	3.93	42.43	47.16	6.48	
Lignite	17.15	44.74	34.89	3.22	0.42
Lignite	1.50	12.16	84.65	2.29	0.70
Rouse Mine	3.13	37.32	30.00	8.25	
Illinois—					
Big Muddy, Jackson Co.	6.40	30.50	54.00	7.50	1.60
Collinsville, Madison Co.	9.20	45.89	31.57	13.34	5.34
Dumferline, Slack, Fulton Co.	9.64	28.86	39.48	22.02	
Gillespie, Macoupin Co.	12.61	30.58	45.27	11.54	1.45
Girard, Macoupin Co.	8.90	32.25	42.89	15.96	8.10
Johnson's, St. Clair Co.	5.50	40.14	40.53	13.83	4.80
Loose's, Sangamon Co.	10.71	37.62	45.07	6.60	2.39
Mercer Co.	8.40	31.20	54.80	5.60	
Mt. Olive, Macoupin Co.	10.38	36.68	46.10	6.84	3.53
St. Clair Co.	7.80	30.69	39.68	21.83	9.62
St. Clair Co.	11.15	34.19	44.94	9.72	4.27
St. John, Perry Co.	9.82	28.35	45.77	16.08	2.06
Streator, La Salle Co.	12.01	35.32	48.78	3.89	2.38
Trenton, Clinton Co.	13.34	30.39	51.96	4.31	0.92
Vulcan Nut, St. Clair Co.	10.30	27.91	48.99	12.80	0.71
Indiana—					
Caking Coal	4.50	45.50	45.50	4.50	
Cannel Coal	3.50	48.00	42.00	6.50	
Indian Territory—					
Atoka	6.66	35.42	51.32	6.60	3.73
Choctaw Nation	1.59	23.31	66.85	8.25	1.18

Name or Locality.	Constituents in per cent of Total Weight.				
	Moisture.	Volatile Matter.	Fixed Carbon.	Ash.	Sulphur.
Iowa—					
Bituminous	5.16	40.21	45.88	8.75	
Kentucky—					
Bituminous	3.60	30.60	58.80	7.00	
Lignite	30.00	23.00	40.00	7.00	
Maryland—					
Cumberland	1.12	16.00	68.19	13.98	0.71
George's Creek	0.74	16.92	76.58	5.86	0.68
Missouri—					
Bituminous	5.06	34.24	47.69	13.01	
New Mexico—					
Bituminous	2.35	35.53	50.24	11.88	0.61
Lignite	10.86	35.14	46.90	7.10	0.64
Ohio—					
Briar Hill, Mahoning Co.	2.47	31.27	64.25	1.45	0.56
Hocking Valley	8.25	35.45	53.15	2.72	0.43
Pennsylvania—					
Cross Creek	3.62	1.96	89.19	5.23	
Lehigh Valley	1.44	7.36	75.20	16.00	
Anthracite Buckwheat	3.88	3.84	81.32	10.96	0.67
Wilkesbarre		7.68	88.90	3.49	
Connellsville	1.93	28.71	63.26	6.10	
Pittsburgh	1.80	35.34	54.94	7.92	1.97
Youghiogheney	2.02	32.14	58.08	6.88	0.88
Tennessee—					
Scott Co.	1.26	35.76	53.14	8.02	1.80
Texas—					
Bituminous	6.67	40.20	43.54	9.59	
Lignite	31.96	39.53	23.05	5.46	1.46
Virginia—					
Kanawha	2.10	34.08	60.67	2.50	0.65
Pocahontas	1.00	21.00	74.39	3.03	0.58
West Virginia—					
Clover Hill	1.34	31.70	56.83	10.13	
New River	0.94	18.19	75.89	4.68	0.30

Coke.

Coke is the solid residuum of coal from which the volatilizable portions have been removed by heat, a process which is illustrated in the action of ordinary furnaces, in which the gasified elements of coal are first burned off, and afterward the fixed or residuary coke. The quantity of coke obtained from coal varies greatly, ranging from 90% to as low as 35%. The average is about 75%.

The quality of the coke obviously depends, in a great measure, on the proportions of the constituent hydrogen and oxygen of the coal from which it is made, which regulate the degree of the fusibility of the coal when exposed to heat.

The average composition of coke is:

Carbon	93.44 per cent.
Sulphur	1.22 per cent.
Ash	5.34 per cent.
	100.00

Coke readily absorbs water, and is capable of absorbing from 15 to 20 per cent of its weight of water. It has been found to absorb as much as 8 per cent of water on its way from the ovens to its destination, in uncovered wagons. Directly exposed to rain, it may absorb 50 per cent of its weight of water. Most of the water is afterward quickly evaporated, leaving from 5 to 10 per cent in the coke. On the average, coke weighs 40 pounds per cubic foot solid.

Wood.

Wood as a combustible may be divided into two classes:

1. The hard, compact, and comparatively *heavy* woods, as oak, beech, elm, ash.

2. The light colored, soft, and comparatively *light* woods, as pine, birch and poplar.

The amount of *moisture* contained in green wood when cut down varies considerably, but on the average it amounts to about 45 per cent of its weight. Wood which has been kept for years in a dry place retains from 15 to 25 per cent of water.

Ordinary firewood is composed of:

Carbon	37.5	per cent.
Hydrogen	4.5	per cent.
Oxygen	30.75	per cent.
Nitrogen	0.75	per cent.
Ash	1.5	per cent.
Hygrometric water	25.0	per cent.
	100.00	

The following table gives the approximate *weight per cord* (128 cubic feet) of various woods (air-dried):

	Weight per Cord. Pounds.
Hickory	4,500
White Oak	3,850
Beech	3,250
Poplar	2,350
Pine (average)	2,000

As to the *heating value* of wood, about 2¼ pounds of dry wood are equal to one pound of average soft coal. The presence of increased quantities of moisture in the wood, detracts of course from its heating value.

In order to obtain the maximum heating value from wood, it is the practice in some factories, as in glass and porcelain works where intensity of heat is required, to dry the wood thoroughly before using it, even employing stoves for this purpose.

Peat, Tan and Straw.

These substances are commercially but little used for fuel in this country, and therefore no data for them are given here.

Petroleum.

The only natural liquid fuel is crude petroleum. Petroleum consists of a number of hydrocarbons, and is found in abundance in the United States (especially in the Ohio Valley, Texas and California), and in Europe, especially in Russia.

Composition of Petroleum.—According to the analyses of M. Saint-Claire Deville, the composition of fifteen petroleums from different sources was found to be practically the same. The

average specific gravity was 0.87 and the average elementary composition was the following:

Carbon	84.7 per cent.
Hydrogen	13.1 per cent.
Oxygen	2.2 per cent.

The total *heating value* of such a petroleum is:

$145 \times (84.7 + 4.28 \times 13.1) = 20{,}411$ B. T. U. per pound.

The *theoretical evaporative power* is:

$0.15 \times (84.7 + 4.28 \times 13.1) = 21.12$ pounds.

Advantages and Disadvantages of Burning Petroleum.—In such parts of the country where coal and labor are high, and oil is low, in price, the latter offers many advantages, among which are the following: The fire can be controlled by a mere turning of a valve. One man can take care of from 8 to 10 large boilers. The heat under the boilers can be kept more uniform, and a more nearly perfect combustion is obtained. There is no handling of ashes, no opening of doors, no fluctuation of heat and no sudden chilling of the boiler. The absence of sulphur in the fuel makes its action on the metal of the boiler less destructive, and the boiler in general remains cleaner externally, and is therefore more effective for heat transmission. Where coal is cheap, however, and where the oil must be shipped over considerable distances, coal fire is by far more economical.

In every day practice about 4 barrels of oil each of 42 gallons, and each weighing 310 pounds, are equal to 1 ton (2,000 pounds) of good bituminous coal, yielding approximately 12,800 B. T. U. per pound. In other words, 2,000 pounds of coal are equal to about 1,240 pounds of oil, or approximately 1.6 pounds of coal correspond to 1 pound of oil. The efficiency obtained in burning oil varies from about 70% to 80%, the average being about 72%. The theoretical evaporative power as given above being 21.12 pounds, the average actual evaporation from and at 212°F. would thus be about 15 pounds per pound of oil.

Method of Burning Petroleum.—In order to maintain the necessary constant pressure, petroleum is usually fed to the burner by a stand pipe. Such a stand pipe is generally about 20 feet high, and often allows heating the crude oil, which is of very thick consistency, in order to make it thinner. The temperature

to which the crude oil is heated is about 145°F. The oil blast is usually directed against a housing of firebrick to obtain complete combustion, and to prevent the intense heat from striking one spot of the boiler continually.

There are many different burners on the market, but not one of them gives entire satisfaction. Often the engineer makes his own burner suited to local conditions. The following cut shows the construction of a burner.

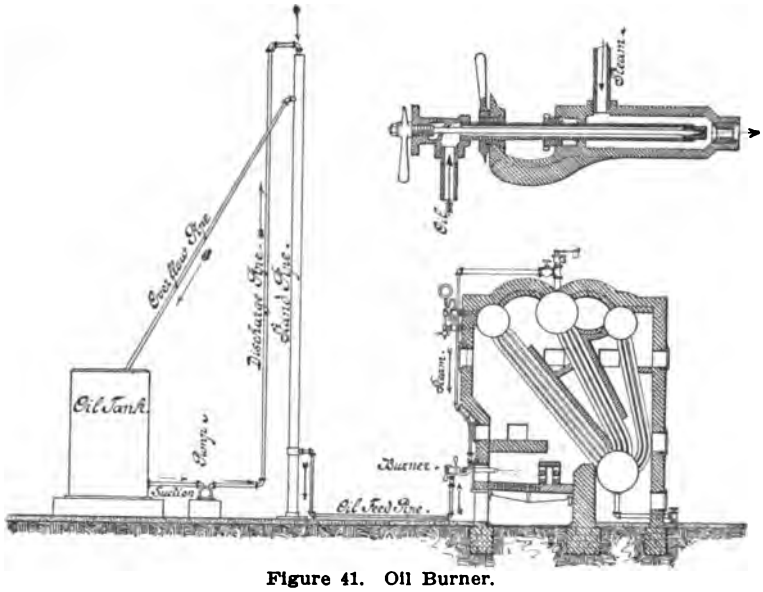


Figure 41. Oil Burner.

Gaseous Fuels.

Gaseous fuels have a great many practical advantages over the solid and even the liquid fuels, and provided their cost is reasonable, may be regarded as ideal fuels. This is true of natural gas, which together with petroleum oil, is found in considerable quantities in New York, Indiana, Pennsylvania and Ohio, as well as in some parts of West Virginia.

Composition of Natural Gas.—Natural gas varies considerably in composition, and consequently also in heating capacity. Not only do the gases from different localities contain their constituents in different percentages, but even if samples of gas are taken from the same well at different times, the analytical re-

Variation in Composition of Natural Gas from the Same Well.*

Constituents.	1	2	3	4	5	6
Marsh gas	57.85	75.16	72.18	65.25	60.70	49.58
Hydrogen	9.64	14.45	20.02	26.16	29.03	35.92
Ethylic hydride	5.20	4.80	3.60	5.50	7.92	12.30
Olefiant gas	0.80	0.60	0.70	0.80	0.98	0.60
Carbon monoxide	1.00	0.30	1.00	0.80	0.58	0.40
Carbon dioxide	0.00	0.30	0.80	0.60	0.00	0.40
Oxygen	2.10	1.20	1.10	0.80	0.78	0.80
Nitrogen	23.41	2.89	0.00	0.00	0.00	0.00

Composition of Natural Gas from Ohio and Indiana.†

Constituents.	Ohio.			Indiana.			
	Fostoria	Findlay	St. Mary's	Muncie	Anderson	Kokomo	Marion
Marsh gas	92.84	93.35	93.85	92.67	93.07	94.16	93.57
Hydrogen	1.89	1.64	1.94	2.35	1.86	1.42	1.20
Olefiant gas	0.20	0.35	0.20	0.25	0.47	0.30	0.15
Carbon monoxide.	0.55	0.41	0.44	0.45	0.73	0.55	0.60
Carbon dioxide...	0.20	0.25	0.23	0.25	0.26	0.29	0.30
Oxygen	0.35	0.39	0.35	0.35	0.42	0.30	0.55
Nitrogen	3.82	3.41	2.98	3.53	3.02	2.80	3.42
Hydrogen Sulphide	0.15	0.20	0.21	0.15	0.15	0.18	0.20

Composition of Different Kinds of Gas in Percent by Weight.*

Constituents.	Natural Gas	Coal Gas	Water Gas	Producer Gas
Marsh gas	90.383	57.20	1.931	1.831
Hydrogen	0.268	8.21	5.431	0.458
Olefiant gas	0.531	10.01	0.000	0.000
Carbon monoxide	0.857	15.02	76.041	25.095
Carbon dioxide	0.700	1.97	10.622	2.517
Oxygen	0.666	1.43	0.965	0.000
Nitrogen	6.178	3.75	3.380	69.413
Water vapor	0.000	2.41	1.630	0.686
Sulphydric acid	0.417	...	....	....
	100.000	100.00	100.000	100.000

* Sturtevant "Mechanical Draft."

† Engineering and Mining Journal.

sults vary widely. These facts are shown by the two tables given on page 153. The first one gives the results of six analyses, made within three months, of a well near Pittsburgh, Pa. The second gives the composition of natural gas from wells in different localities.

In the table on the composition of different kinds of gas, page 153, the natural gas is that of Findlay, Ohio. The coal gas is an average sample of gas purified for use as an illuminant. The water gas sample represents gas for heating purposes, and consequently is not purified; hence the higher percentage of carbon dioxide it contains. The producer gas is an average sample taken at a steel works and is made from anthracite, and is therefore not of such a high grade as would be the case, had it been made from soft coal.

The following comparison of the value of the different gases, the analyses of which were given above, shows the quantity of water evaporated from and at 212°F. by 1,000 cubic feet of each gas, allowing an excess of 20% of air and permitting the resulting gases to escape at a temperature of 500° F., the air for combustion entering at 60°F.

Pounds of Water Actually Evaporated from and at 212° F. by 1,000 Cubic Feet of Different Gases.

Gas.	Water evaporated.
Natural Gas	893 pounds.
Coal Gas	591 pounds.
Water Gas	262 pounds.
Producer Gas	115 pounds.

Heating Value of Natural Gas.—Experiments made in Pittsburgh with natural gas have shown that in actual efficiency 1,000 cubic feet of gas are equivalent under the steam boiler to from 80 to 130 pounds of coal, or approximately to 100 pounds of coal. Under favorable conditions, one cubic foot of gas may evaporate approximately one pound of water from and at 212°F.

Advantages and Disadvantages of Burning Gas.—The convenience and economy attending the use of natural gas have led many people to believe that fuel gas made from coal at large central stations and distributed to factories, etc., would be the fuel of the future. This is erroneous, however, unless the gas be used directly for producing power, as for instance, in a gas engine. In

the latter case, the latest constructions are claimed by their inventors to use from 1 to 1.5 pounds of coal, or from 1.5 to 2 pounds of coke made into gas, per horse-power per hour.

The following calculation will show that it is not economical to convert bituminous coal into gas and distribute it to boiler plants.

In a well designed fuel gas plant, employing an average Illinois coal, the gas obtained from one ton of the coal, furnished 10,105,480 B. T. U., or one pound 5,052. 7 B. T. U. A calorimetric test, made directly with the coal, showed it to yield 11,172 B. T. U., so that by converting the coal into gas and burning the latter, a loss of 11,172 — 5,052.7 or 6,119.3 B. T. U. was sustained. This corresponds to a loss in heat of 54.8%.

FURNACES.

The furnace or fire-box of a boiler is an apparatus for generating heat by the burning of fuel. It is often provided with an extension, the combustion chamber, the object of which is to insure complete combustion of the gases which are distilled from the burning fuel.

Requirements of a Good Furnace.

A furnace should be able to create a high temperature while burning as little fuel as possible and while employing the minimum quantity of air necessary for combustion. It should be cheap and durable and require little labor for operation.

Location of Furnace.

The furnace may be located as follows:

Inside of the Boiler.—This occurs, for instance, in the locomotive boiler, the Scotch Marine boiler and the vertical tubular boilers.

Under the Boiler.—This is the usual place for the furnace, and with most coals and furnaces gives good results, as the grate surface as well as the combustion chamber can be made amply large.

In Front of the Boiler.—With this arrangement the furnace is absolutely separate from the boiler, and only the hot combustion gases come in contact with the latter. It gives very good

results when used on inclined water-tube or vertical water-tube boilers. These furnaces are often called *Dutch ovens.*

While in furnaces in general the air enters below the grate and passes upward through the incandescent coal, this current of air may also be reversed and to good advantage. This is done in the *Hawley Down-Draft furnace* which is illustrated by Figure 42. This furnace, as will be seen, is provided with two grates located one above the other. The lower grate is an ordinary one and is not fired, but receives its supply from the grate above, the latter allowing the coked coal to fall through. The upper grate

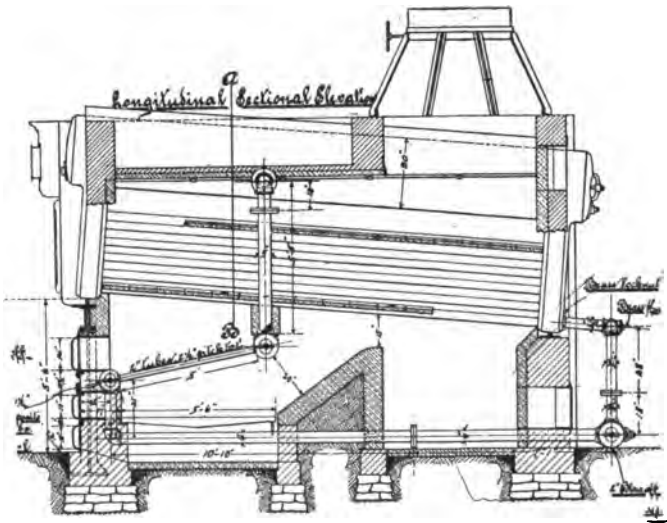


Figure 42. Hawley Down-Draft Furnace.

is formed by a series of water pipes running lengthwise and connected to headers at their rear and front ends. The front header receives the water from the boiler, while the rear header returns it, causing a rapid circulation of the water in the boiler. The coal is fed on the upper grate and the gases produced are forced to pass downward by a wall erected at the place where otherwise the bridge would be. This wall stops the progress of the gases and leads them through the coal downward to the lower grate, where they meet the coked fuel, so that the smoke coming from the upper furnace is burned. The gases then flow as usual into the flues, towards the chimney. The upper grates are in-

clined towards the front about three inches per foot of length, and the lower grates inclined towards the rear about two inches per foot.

Grates.

We distinguish between the *common* and the *shaking* grate. In the former the bars are stationary and are either single or double, while in the shaking grate the bars are made movable. This is done for the reason that during the coking process, the coal bed has a tendency to form a solid crust, which must be

Figure 43. Shaking Grate.

broken up in order to allow the air to pass through freely. In this respect the shaking grate has the advantage over the common grate, inasmuch as the door need not be opened. Furthermore, shaking of the grate at the same time partly cleans the fire, which operation with the common grate necessitates opening of the door.

Figure 43 shows one of the numerous types of shaking grates on the market.

Air Passages through Grate Bars.—The usual practice is to make the total air opening equal to 30% to 50% of the area of the grate, the larger the better in fact, especially when the coal contains a large amount of ashes. The distance from grate to boiler shell should be governed by the kind of coal to be burned. Anthracite coal requires from 20 to 24 inches for the best results; for semi-bituminous coal the distance should be between 26 and

30 inches; and for bituminous coal, rich in volatile matter, not less than 36 inches.

Methods of Firing by Hand.

In order to burn the fuel as completely as possible and thus obtain the greatest possible amount of heat from it, attention must be given to feeding the fuel and keeping it in a proper state in the furnace. In firing by hand, the three following methods may be distinguished:

1. Spreading the coal evenly over the whole surface.
2. Alternating by filling one-half of the furnace at a time.
3. Coking the fuel by banking it in front, and the next time spreading it and again banking fresh fuel in front.

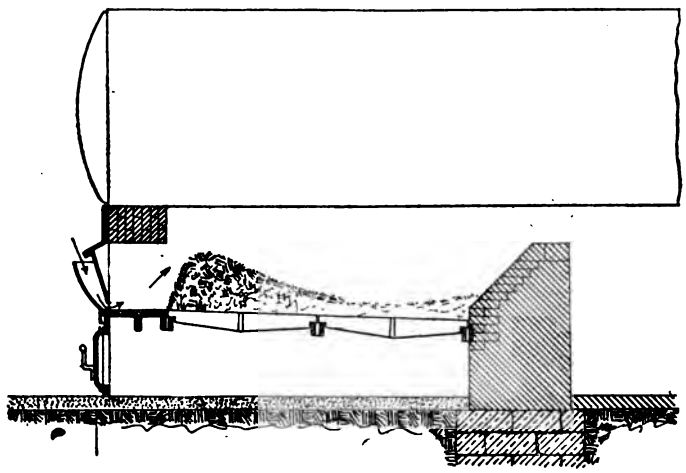


Figure 44. Coking Method of Firing.

To properly apply the *spreading method,* the fireman should wait until the volatile matter from the last charge of coal is entirely distilled off, leaving a hot fire-bed of burning coal, before adding a new charge of coal. This must be sprinkled evenly over the surface of the fire, so as not to choke the openings in the porous coke bed through which the air passes. In this way, economical and comparatively smokeless results can be obtained. However, firemen who properly handle this method of firing are very rare, and engineers should therefore study the proper methods of

firing, so as to be able to instruct their firemen. Anthracite coal should always be burned by the spreading method.

In *alternate firing,* a somewhat easier method is followed by the fireman, in that the coal is thrown to one side of the grate only. The result is, that the excess of air flowing through one-half of the bed of glowing coke will furnish the highly heated oxygen required to complete the combustion of the volatile gases distilled from the coal on the other half of the bed. After this charge of coal has its volatile matter distilled off, the fireman similarly charges a supply of coal upon the other half of the fire-bed, and afterwards continues stoking, first on one side and then on the other, which system of firing gives this method its name of alternate firing.

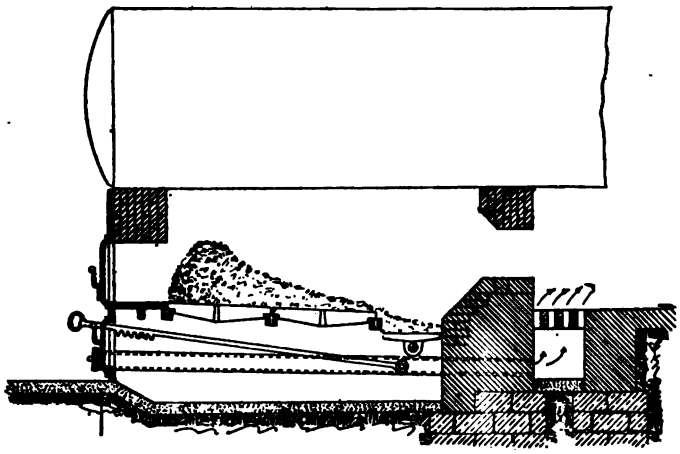


Figure 45. Coking Method of Firing.

To describe the *coking method* of firing, let us refer to Figure 62, and let us suppose that the last charge of bituminous coal has had most of its volatile gases distilled off, so that the grate bars are covered with a bed of incandescent coke. In charging fresh fuel, the coal should be piled up just inside of the door, allowing the coal to occupy from one-quarter to one-third of the grate's length. A slow distillation of the volatile products from the fresh pile of coal then takes place, and the resulting products, principally hydrocarbon gases, pass over the incandescent bed of coke covering the rear of the furnace, where they meet an excess

of air heated to a high temperature during its passage through the furnace. The air and the volatile combustible gases are thus mixed at a high temperature, causing a rapid combustion, which is a most favorable condition for fuel economy and smoke prevention. Whenever it becomes necessary to put another charge of coal into the furnace, the fireman pushes the previous charge to the rear end of the grates, where he spreads it evenly, and then places a fresh charge of coal in front. After the fireman becomes familiar with this method of firing and also familiarizes himself with the action of the coal he handles, then, in a properly designed furnace, highly economical and practically smokeless results may

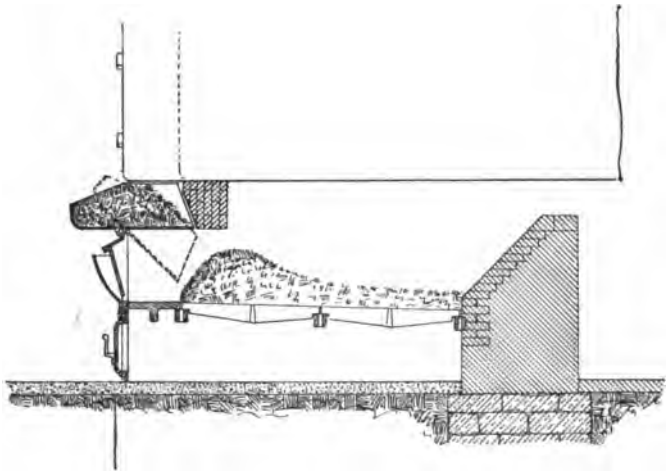


Figure 46. Improved Coking Method of Firing.

be obtained with no more difficulty than by any other method of hand-firing. Figures 44 and 45 illustrate the coking method of firing.

Figure 46 shows an *improvement in the coking method* of firing. The coal charging hopper is placed over the firing doors of the furnace and extends from one side wall to the other. After the fireman shoves the coal back from the front of the furnace, a lever is operated which dumps the hopper as shown by the dotted line, thus placing the coal in the desired position in the furnace. The hopper is then tilted back in its original position and refilled with coal. Figure 45 shows that the coal and the refuse can be dumped.

In slow burning furnaces with long flues, *moistening the fuel* before throwing it in, and moistening the ash pit, produce better results from the fuel. The heat radiating upon the ashes produces steam. Steam lessens the glow fire or flameless incandescence of the fuel, and increases the quantity of the flame by forming carbon monoxide and hydrogen gases in its decomposition into its elements, and the reduction by the oxygen of the carbon dioxide already formed in the furnace. The gases thus produced are afterward burned in the flues. The presence of moisture even in coke gives rise to a flame in the flues and reduces the intensity of the heat in the glow fire. It aids in distributing combustion over a larger space.

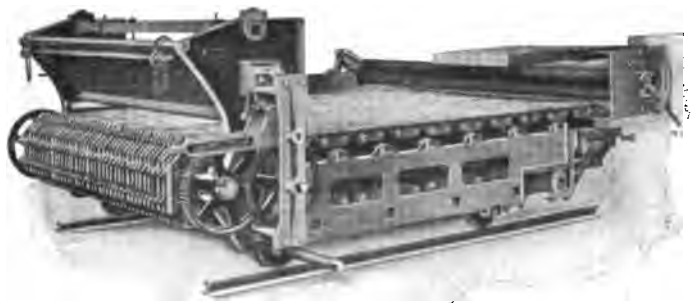


Figure 47. Chain Grate.

Mechanical Stokers.

Mechanical stokers are devices by means of which coal is automatically and continually fed to the furnace. Their object is to save manual labor, to avoid the frequent opening of the fire doors which always causes cooling of the fire, and to insure uniform and complete combustion, air and coal supply being continuous and regular.

For large boilers or for a battery or number of boilers, mechanical stokers are a good investment, especially when connected with the proper coal and ash handling machinery. One fireman is then able to take care of boilers aggregating up to 4,000 horsepower capacity, while with hand-firing there must be one fireman for about every 400 boiler horse-power.

For smaller plants the interest on the investment makes hand-firing the less expensive of the two systems. Still, mechanical

stokers are found in plants as small as 200 horse-power capacity, when for instance the fuel is of poor grade and it is desirable to have as little smoke as possible.

Chain Grate.—One type of this stoker is illustrated by Figure 47. The grate bars are supported by endless chains, which run on sprocket wheels driven by gearing. The entire grate and driving mechanism are mounted on a permanent truck, which can easily be moved out of the furnace. The coal is fed into the hopper at the front end, preferably from a large, elevated hop-

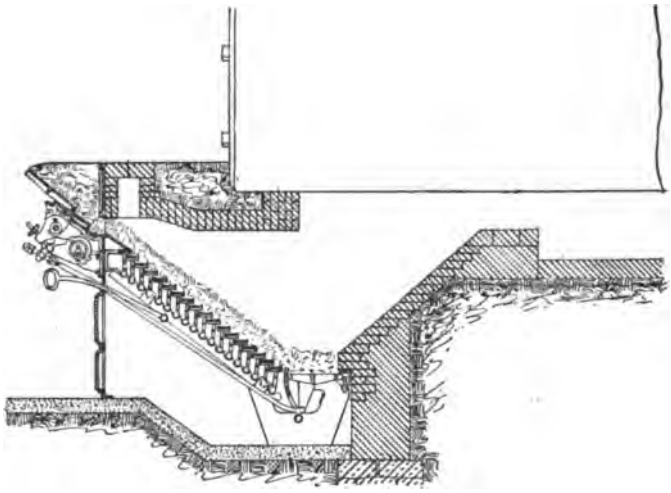


Figure 48. Roney Stoker.

per, and a gate regulates the height of the coal layer on the grate. The continuous slow motion of the chains insures a uniform coal supply to the furnace. The speed, and the height of the layer of fuel should correspond in such a way that the fuel has all burned to ashes when it reaches the rear of the furnace. The ashes are then discharged into the pit, and no cleaning of the grate is necessary.

Step Grates.—Figure 48 shows a mechanical stoker of this type. Instead of the actual motion of the grate bars in the chain grate, the bars are here subjected only to a reciprocating motion. Each grate bar rocks through an arc of 30 degrees, assuming alternately horizontal and inclined positions, thus pushing the coal

forward. The coal is fed into the hopper at the front end, and from there is forced to the grate by means of the reciprocating pusher. At the bottom of the incline there is a dumping grate for receiving and discharging ash and clinkers.

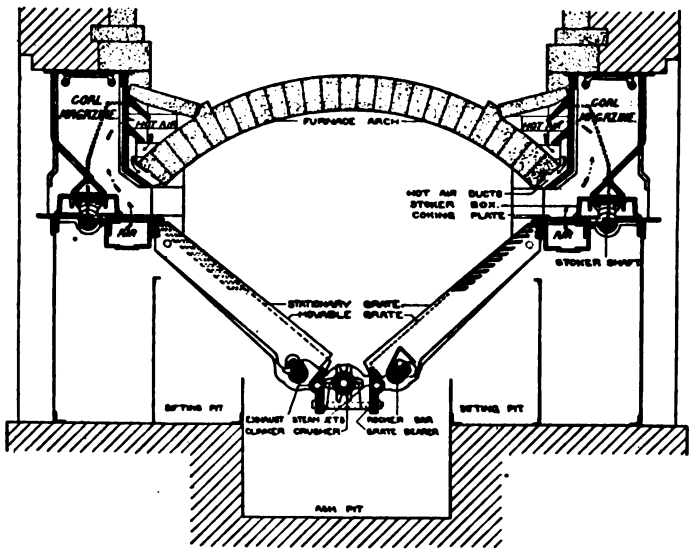


Figure 49. Murphy Furnace.

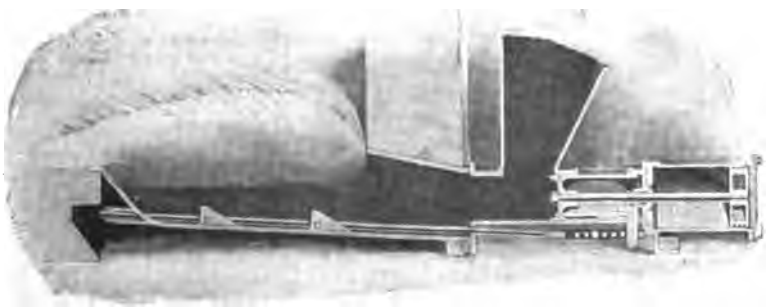


Figure 50. Underfeed Stoker.

The stoker shown in Figure 48 has a front feed. Other stokers of this type are fed from the sides, the two inclines meeting at the center of the furnace (see Figure 49).

Underfeed Stokers.—Figure 50 shows one design of this type of mechanical stoker. There are no grate bars. The combustion takes place in a cast iron retort, in the side of which tuyere blocks are provided, for the admission of air to the fuel. The coal is fed to the hopper in front, pushed into the retort by a piston-like ram and two auxiliary rams, and forced upward underneath the fire, replacing the incandescent coal and ashes as

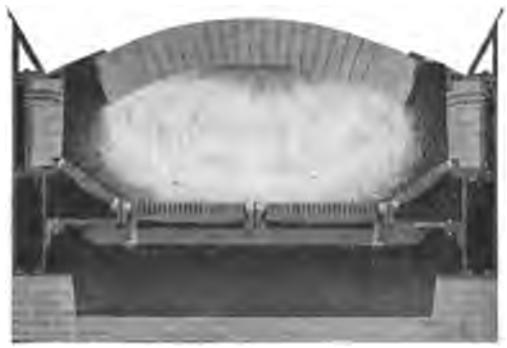


Figure 51. Dutch Oven Gravity Feed Furnace.

these slide out on the dead plates on the side. The ashes are raked off from the dead plates by hand. The air is supplied by a blower and admitted into the fuel just below the fire. The rams are driven from a steam cylinder, and the speed may be regulated either automatically or by hand. Another design of this class of stoker has a screw conveyor instead of the rams, to force the fuel into retort.

Dutch Oven Gravity Feed Furnaces.—Figure 51 shows the so-called Burke Furnace, which is a dutch oven built partly in front of the boiler in which the coal is fed from the top. This is a coking type furnace in which the coal can be burned practically smokeless and with high efficiency.

CHIMNEYS.

Chimneys may be constructed of brick, steel, or reenforced concrete. They are usually made round, such being preferable because they offer less resistance to the wind, do not retard the revolving action of the gases, and, having the smallest circumference for a given cross-section, suffer the least from loss of heat. The bottom of the opening of the chimney should be at least 24 to 30 inches from the bottom of the bridging. If more than one bridging enters the chimney, these should be so arranged as to give the gases from all of them the same direction, since otherwise a considerable loss of velocity would occur.

Brick chimneys are the best, but also the most expensive. They last long and need neither painting, nor frequent repairs, except pointing out once in a long while. They retain the heat much better than steel smokestacks.

So-called **radial brick** is also used with good results in chimneys, also the reenforced concrete tile chimneys.

Concrete reenforced by iron bars has come into extensive use of late for chimneys. Such chimneys occupy less space than the brick chimney, and possess great resistance. They have not always been successful, due to either faulty construction or poor material.

Steel smokestacks are either made self-sustaining or are held by guy ropes. The self-sustaining stack is provided with a strong cast-iron base, bolted upon a foundation heavy enough to prevent the strongest wind from overturning the stack. Guy ropes are used in the smaller sizes. Steel chimneys have several advantages, among them being light weight, small space occupied, and low cost. The necessity of frequent painting, and the danger of rusting on the inside, if not continually in use, are two drawbacks.

Chimney Draft.

The draft of a chimney depends on the height of the chimney and on the difference between the temperature of the outside air and that of the chimney gases. The draft is most effective when the ratio between absolute inside temperature and absolute outside temperature is 25 to 12, which would be the case, for in-

stance, if the outside temperature were 20° F. and the inside temperature 540° F. Thus:

$$\frac{460 + 20}{460 + 540} = \frac{480}{1{,}000} = \frac{12}{25}$$

Calculation of Draft.—If the height of the chimney, the temperature of the outside air, and the temperature of the chimney gases are known, the draft theoretically produced by the chimney may be calculated. The method given below does not take friction into account, and to approximate the actual draft, about 10 per cent may be deducted.

Example.—What draft will a chimney 100 feet high theoretically produce, if the temperature of the outside air is 62°F. and that of the flue gases is 475°F.

Solution 1.—From the table, page 167, one cubic foot of gas at 475° F. weighs approximately 0.0447 pound. A column of gas 100 feet high with a base of 1 square foot has a volume of 100 cubic feet, and will weigh $100 \times 0.0447 = 4.47$ pounds. The column therefore exerts a pressure of 4.47 pounds per square foot. From the same table, a cubic foot of air at 62°F. is found to weigh approximately 0.0761 pound. A column 100 feet high and 1 square foot in section will therefore weigh $100 \times 0.0761 = 7.61$ pounds. The difference in pressure is consequently $7.61 - 4.47 = 3.14$ pounds per square foot.

Since the draft is equal to the difference in pressures, but is expressed in inches of water column (see Measurement of Draft, page 167), it may be calculated from this as follows:

$$\text{Draft in inches of water} = \frac{3.14 \times 12}{144 \times 0.433} = 0.604 \text{ inch.}$$

The figure 0.433 represents the pressure per square inch which sustains a column of water 12 inches high at 62°F., and is obtained by dividing 62.35, the weight in pounds of one cubic foot of water of 62°F., by 144.

Solution 2.—The above calculation may be combined into a formula, which gives practically the same results, thus:

$$D = H \times \left(\frac{7.6}{T} - \frac{7.9}{T_1}\right)$$

D = draft in inches.

H = height of chimney in feet.
T = absolute temperature in degrees F. of outside air.
T_1 = absolute temperature in degrees F. of chimney gases.
By inserting the figures in this formula we obtain:

$$D = 100 \times \left(\frac{7.6}{62 + 460} - \frac{7.9}{475 + 460} \right) = 0.611 \text{ inch.}$$

From the above it may be seen that the higher the smoke stack the lower need be the temperature of the gases to obtain the same draft.

Density of Air and Chimney Gases at Various Temperatures.

AIR		CHIMNEY GASES					
Temp. Degrees F.	Pounds per Cubic Foot	Temp. Degrees F.	Pounds per Cubic Foot	Temp. Degrees F.	Pounds per Cubic Foot	Temp. Degrees F.	Pounds per Cubic Foot
0	.086353	200	.06334	430	.04695	660	.03730
5	.085424	210	.06239	440	.04643	670	.03697
10	.084513	220	.06147	450	.04592	680	.03665
15	.083623	230	.06058	460	.04542	690	.03633
20	.082750	240	.05971	470	.04493	700	.03602
25	.081895	250	.05887	480	.04445	710	.03571
30	.081058	260	.05805	490	.04398	720	.03540
32	.080728	270	.05726	500	.04353	730	.03511
35	.080238	280	.05648	510	.04308	740	.03481
40	.079434	290	.05573	520	.04264	750	.03453
45	.078646	300	.05499	530	.04221	760	.03424
50	.077874	310	.05428	540	.04178	770	.03396
55	.077117	320	.05358	550	.04137	780	.03369
60	.076374	330	.05290	560	.04096	790	.03342
62	.076081	340	.05224	570	.04056	800	.03316
65	.075645	350	.05159	580	.04017	900	.03072
70	.074930	360	.05096	590	.03979	1000	.02861
75	.074229	370	.05035	600	.03942	1100	.02678
80	.073541	380	.04975	610	.03905	1200	.02516
85	.072865	390	.04916	620	.03869	1300	.02373
90	.072201	400	.04859	630	.03833	1400	.02245
95	.071550	410	.04803	640	.03798	1500	.02131
100	.070910	420	.04749	650	.03764	1800	.01848
110	.069665	...	.....	...	.....	2000	.01698

Measurement of Draft.—The ordinary draft of a chimney is ½ inch to 1 inch for ordinary heights and temperatures.

The difference in pressure may be measured by the height of the column of water whose weight or head is equal to the difference in pressures. For such a test, a U-shaped tube, with one opening connected to the inside of the chimney and the other end open to the air, is used. This tube is party filled with water,

and a wooden scale having graduations ⅛ inch apart, is placed between the two tubes. As the pressure of the atmosphere outside is greater than that of the gases inside the chimney, this excess pressure will force the water downwards in the outer tube and thereby raise it a corresponding distance in the inner tube. The pressure or draft is measured by taking the difference in height between the two water columns, being usually expressed in inches. By a ¾-inch draft, for instance, is meant that a column of water can be raised to a height so that the difference in level between the two columns is ¾-inch, this being effected by the difference in weight between the air and the chimney gases. Such a simple draft gauge is illustrated by Figure 52.

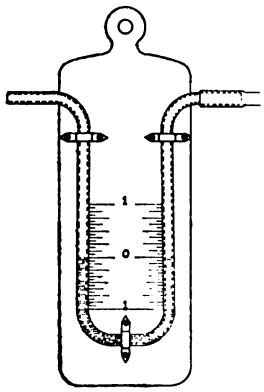


Figure 52. Simple Draft Gauge.

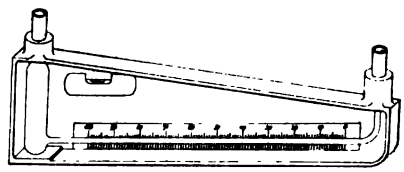


Figure 53. Ellison Draft Gauge.

The *Ellison draft gauge* is more accurate than the one described above, since it allows reading to the 100th part of an inch. Figure 53 shows its construction.

Area and Diameter of Chimneys.

On account of retardation of the gases due to friction, the *effective area* of the chimney is less than the *actual area.* The approximate formula for the effective area of either square or round chimney is:

$$E = A - 0.6 \times \sqrt{A}.$$

E = effective area.

A = actual area.

If the actual area, for instance, is 6.361 square feet, then the effective area $= 6.361 - 0.6 \times \sqrt{6.361} = 4.848$ square feet.

There is a corresponding difference between the actual and effective diameters of a round chimney, and the actual and effective sides of a square chimney. In round chimneys the actual diameter is found by adding 4 inches to the effective diameter, or if the effective area E is known in square feet, the *actual diameter* is found in inches as follows:

$$d = 12 \times 2 \times \sqrt{\frac{E}{3.1416}} + 4 = \frac{24}{1.7724} \times \sqrt{E} + 4$$

$$= 13.54 \times \sqrt{E} + 4.$$

In the square chimneys, the actual side equals the side of the effective area plus 4 inches, or if the effective area E is known in square feet, the side of the chimney in inches is

$$s = 12 \times \sqrt{E} + 4.$$

The effective area in square feet and the actual diameter in inches of a round chimney of a given height, which should produce a given horse-power, can be found according to the following formula, obtained by transposing the formula for horse-power given on page 171.

$$E = \frac{0.3 \times H.P.}{\sqrt{H}}$$

E = effective area in square feet.
H. P. = horse-power required.
H = height of chimney in feet.

From the value of the effective area here found, the actual diameter may be calculated according to the formula given above.

Example.—What must be the effective area and actual diameter of a chimney which should be 130 feet high and produce 450 horse-power?

Solution.—$E = \frac{0.3 \times 450}{\sqrt{130}} = 11.85$ square feet.

$d = 13.54 \times \sqrt{11.85} + 4 = 51$ inches.

The effective area is 11.85 square feet, and the actual diameter is 51 inches.

The actual diameter of a chimney for a certain horse-power and height may also be found by the following formula, which

is simply a combination of the above two for effective area and diameter.

$$d = 13.54 \times \sqrt{\frac{\text{H. P.} \times 0.3}{\sqrt{H}}} + 4$$

Height of Chimney.

If the number of horse-power delivered and the effective area are known, the height of a chimney may be calculated according to the following formula, which is derived from the horse-power formula, page 171, by transposition.

$$H = \left(\frac{0.3 \times \text{H. P.}}{E}\right)^2$$

H = height of chimney in feet.

H. P. = horse-power delivered by chimney.

E = effective area in square feet.

Example.—What must be the height of a chimney whose effective area is 4.845 square feet, if it should deliver 161.5 H. P.

Solution.—$H = \left(\frac{0.3 \times 161.5}{4.845}\right)^2 = \left(\frac{48.45}{4.845}\right)^2 = 100$ feet.

The chimney must be 100 feet high.

The height of a chimney can also be found if the draft is given, and the temperatures of the outside air and the chimney gases are known.

Example.—Assuming the temperature of the air to be 62° F. and the temperature of the chimney gases 475° F., what must be the height of a chimney to create a draft of 0.604 inch?

Solution.—From the table, page 167, it is found that 1 cubic foot of air of 62° F. weighs 0.0761 pound, and that 1 cubic foot of gas of 475° F. weighs 0.0447 pound. The difference between the weight of one cubic foot of air at 62° F. and one cubic foot of hot chimney gas at a temperature of 475° F. is, therefore, 0.0761 — 0.0447 = 0.0314 pound. In other words, for a height of 1 foot, the difference in pressure amounts to 0.0314 pound per square foot.

A draft of 0.604 inch is equal to $\frac{0.604 \times 144 \times 0.433}{12}$ or approximately 3.14 pounds of pressure per square foot.

The figure 0.433, as explained under draft, is the pressure per square inch which sustains a column of water 12 inches high at 62° F.

Since a difference in pressure of 0.0314 pound corresponds to a height of one foot, a difference of 3.14 pounds must correspond to a height of $\frac{3.14}{0.0314}$ or 100 feet. Hence the chimney must be 100 feet high.

A result practically the same as the above may be obtained by using the following formula, which combines the above calculations. The formula may be obtained by transposing the one used for calculating draft (see page 166).

$$H = \frac{D}{\frac{7.6}{T} - \frac{7.9}{T_1}}$$

H = height of chimney in feet.

D = draft in inches.

T = absolute temperature in degrees F. of boiler room.

T_1 = absolute temperature in degrees F. of gases entering chimney.

Example.—How high must a chimney be to produce 0.604 inch draft, the outside temperature being 62° F., and the chimney temperature 475° F.?

Solution.—$H = \frac{0.604}{\frac{7.6}{62+460} - \frac{7.9}{475+460}} = 98.8$, or approximately 100 feet.

Horse-Power Capacity of Chimney.

By horse-power capacity of a chimney is meant the boiler horse-power which the chimney of given size is capable of developing. It varies directly as the square root of the height, and is in direct proportion to the effective area. It may be calculated by the following formula:

$$\text{H. P.} = 3.33 \times E \times \sqrt{H}$$
$$= 3.33 \times (A - 0.6 \times \sqrt{A}) \times \sqrt{H}$$

A = cross-sectional area of the chimney in square feet.

E = effective area in square feet.

H = height of the chimney in feet.

Example.—How many H. P. can a chimney of 916 square inches (6.361 square feet) cross-section and 100 feet height deliver?

Solution.—H. P. $= 3.33 \times (6.361 - 0.6 \times \sqrt{6.361}) \times \sqrt{100}$ $= 161.5$ H. P.

By transposing the formula for diameter given on page 170, the following formula for horse-power in terms of diameter and height is obtained:

$$\text{H. P.} = \frac{(d-4)^2 \times \sqrt{H}}{0.3 \times (13.54)^2}$$

Example.—The actual diameter of a chimney is 33½ inches, and the height 100 feet. How many horse-power can it deliver?

Size of Chimneys for Steam Boilers.

Diameter, Inches	Area (A) Sq. Ft.	Effective Area E=A−0.6√A Square Feet.	HEIGHT OF CHIMNEY. Commercial Horse-Power of Boiler. 50 ft.	60 ft.	70 ft.	80 ft.	90 ft.	100 ft.	110 ft.	125 ft.	150 ft.	175 ft.	200 ft.	225 ft.	250 ft.	300 ft.	Equivalent Sq. Chimney. Side of Sq. √E + 4 Inches
18	1.77	.97	23	25	27	29	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	16
21	2.41	1.47	35	38	41	44	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	19
24	3.14	2.08	49	54	58	62	66	.....	.....	.....	.....	.....	.....	.....	.....	.....	22
27	3.98	2.78	65	72	78	83	88	.....	.....	.....	.....	.....	.....	.....	.....	.....	24
30	4.91	3.58	84	92	100	107	113	119	.....	.....	.....	.....	.....	.....	.....	.....	27
33	5.94	4.48	.....	115	125	133	141	149	156	.....	.....	.....	.....	.....	.....	.....	30
36	7.07	5.47	.....	141	152	163	173	182	191	204	.....	.....	.....	.....	.....	.....	32
39	8.30	6.57	.....	.....	183	196	208	219	229	245	.....	.....	.....	.....	.....	.....	35
42	9.62	7.76	.....	.....	216	231	245	259	271	289	316	.....	.....	.....	.....	.....	38
48	12.57	10.44	.....	.....	.....	311	330	348	365	389	426	.....	.....	.....	.....	.....	43
54	15.90	13.51	.....	.....	.....	.....	427	449	472	503	551	595	.....	.....	.....	.....	48
60	19.64	16.98	.....	.....	.....	.....	536	565	593	632	692	748	.....	.....	.....	.....	54
66	23.76	20.83	.....	.....	.....	.....	.....	694	728	776	849	918	981	.....	.....	.....	59
72	28.27	25.08	.....	.....	.....	.....	.....	835	876	934	1,023	1,105	1,181	1,263	.....	.....	64
78	33.18	29.73	.....	.....	.....	.....	.....	.....	1,038	1,107	1,212	1,310	1,400	1,485	1,565	.....	70
84	38.48	34.76	.....	.....	.....	.....	.....	.....	1,214	1,294	1,418	1,531	1,637	1,736	1,830	2,005	75
90	44.18	40.19	.....	.....	.....	.....	.....	.....	.....	1,490	1,639	1,770	1,893	2,008	2,116	2,318	80
96	50.27	46.01	.....	.....	.....	.....	.....	.....	.....	1,712	1,876	2,027	2,167	2,298	2,423	2,654	86
102	56.75	52.23	.....	.....	.....	.....	.....	.....	.....	1,944	2,130	2,300	2,459	2,609	2,750	3,012	91
108	63.69	58.53	.....	.....	.....	.....	.....	.....	.....	2,090	2,399	2,592	2,771	2,939	3,098	3,393	96
114	70.88	65.83	.....	.....	.....	.....	.....	.....	.....	.....	2,685	2,900	3,100	3,288	3,466	3,797	101
120	78.54	73.22	.....	.....	.....	.....	.....	.....	.....	.....	2,966	3,220	3,448	3,657	3,855	4,223	107
132	95.03	89.18	.....	.....	.....	.....	.....	.....	.....	.....	3,637	3,999	4,300	4,455	4,696	5,144	117
144	113.10	106.72	.....	.....	.....	.....	.....	.....	.....	.....	4,352	4,701	5,026	5,331	5,618	6,155	126

Solution.—

$$\text{H. P.} = \frac{(33.5-4)^2 \times \sqrt{100}}{0.3 \times (13.54)^2} = \frac{29.5 \times 29.5 \times 10}{0.3 \times 13.54 \times 13.54} = \frac{8{,}702}{55} = 158.2.$$

Instead of using the formulas given on the preceding pages for the relationship between effective area or actual diameter, height and horse-power, the table on page 172, which is calculated from the formula for horse-power, may be employed. The figures are based on the assumption that 5 pounds of coal are required per boiler horse-power, this being a liberal allowance.

Velocity of Gases in Chimney.

The velocity of the gases in a chimney varies directly as the square root of the height. It is also materially influenced by temperature, for the higher the inside, and the lower the outside temperature, the greater is the difference in pressure between the two columns of air, and the greater the difference in pressure the greater is the velocity of the gases going up the chimney. The volume of gases a chimney can discharge is, therefore, in direct proportion to the velocity, but as the density of the gases decreases with the temperature, a point can be reached where as much is lost in weight of the gas passing through the chimney as is gained by the increased velocity.

The mean velocity of the gases in chimneys can be found according to the following formula:

$$V = \frac{8 \times \sqrt{H} \times (t_1 - t)}{3.33 \times t_1}$$

V = mean velocity of chimney gases in feet per second.

H = height of chimney in feet.

t_1 = temperature of chimney gases in degrees F.

t = temperature of air in boiler room in degrees F.

Example.—A chimney is 130 feet high, temperature of chimney gases 470° F., and temperature of boiler-room, 70° F. What will be the velocity of the chimney gases?

Solution. — $V = \frac{8 \times \sqrt{130} \times (470-70)}{3.33 \times 470} = 23.3$ feet per second, or approximately 16 miles per hour.

Steam Boilers

Generally speaking, a *steam boiler* is a closed metallic vessel both strong and tight, in which steam is generated from water by the application of heat, for the purpose of giving motion to machinery or for supplying heat in a convenient manner to places where it is needed.

To make a boiler complete requires a number of *fixtures, appendages* and *accessories*. There must be a steam gauge, safety valve, water column with gauge glass and gauge cocks to show the height of water in the boiler (water level), a feed pump or injector with a supply pipe, feed valve, check valve, a blow-off pipe with a blow-off valve inserted for the purpose of lowering the height of water in the boiler or to empty it entirely, a heater to bring the temperature of the feed-water as nearly as possible to that of the boiler water, a steam pipe with stop valve to convey the steam to the place where it is wanted, a reverse valve to prevent the formation of a vacuum in the boiler, manholes and hand holes with their guards, a fusible plug, a high and low water alarm, and in some cases there is a superheater.

All these are attachments to the boiler proper, and have direct reference to its internal functions, but in addition there are a number of *external appendages* and belongings, such as the brackets or lugs, pedestals and other items for the support of the boiler, the masonry in which it is set, with its binders, buckstaves, buckstave rods, wall plates, anchor rods, the boiler front with its doors, i. e. fire doors, ash doors, flue caps, arch plates, bearer bars, grate bars, side and rear cleaning doors, dampers and chimney. These are all equally necessary to enable the boiler to perform its duty properly. Furthermore, there are observing and working instruments such as a draft gauge, a feed-water thermometer, flue gas thermometer, means for weighing the coal

and the feed-water in order to determine the efficiency of the boiler and furnace, fire tools, flue brushes, scrapers and blowers, scaling tools and a hose for washing out the boiler, also hammers, chisels, wrenches, etc.

Thus we see in speaking of a boiler, not only the boiler proper is meant, but also its fixtures, appendages and belongings, all of which must be kept in good condition at all times, in order to insure safety, and obtain the best results possible pertaining to the generation of steam.

TYPES OF STEAM BOILERS.

According to their form and construction, boilers may be classified as follows: horizontal and vertical; internally and externally fired; fire-tube and water-tube. A fire-tube boiler is one in which the combustion gases pass through the tubes, and a water-tube boiler is one in which the water passes through the tubes. Combining these features in various ways allows a large number of different designs. According to their use boilers may be classified as stationary, locomotive, marine, and portable boilers.

The internally fired types more generally in use in this country are the Scotch marine boiler, the locomotive or fire-box boiler and the upright tubular boiler. Of the externally fired type, those most generally in use are the horizontal return tubular boiler, the vertical and horizontal water-tube boilers, and combinations of both.

Scotch Marine Boiler.

This boiler is of the horizontal and fire-tube type, internally fired. As the name indicates, it is mainly used on steamships, but in the somewhat modified design shown in Figure 54, it is also being used to some extent for stationary work.

The special feature of this boiler is, that a cylindrical flue of corrugated steel, large enough to accommodate grate, fuel and ash pit, extends through the full length of the boiler. To the front of the boiler is riveted the uptake, and to the rear end the combustion chamber, which is lined with fire brick. The combustion gases pass through the flue to the combustion chamber,

return to the front end through the fire tubes, and from the uptake pass to the chimney.

The arrangement for obtaining dry steam shown in the cut, in connection with the steam outlet, is not a feature of this boiler, but may be used in any type of boiler. A perforated pipe extends along the top of the boiler, the perforations being on the upper side of the pipe only, and through these the outgoing steam has to pass.

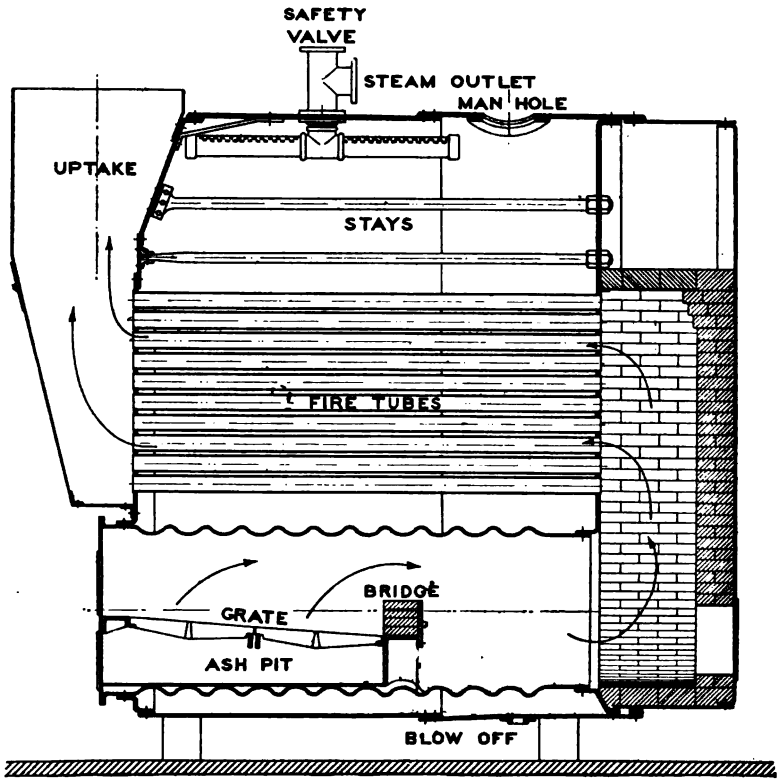


Figure 54. Scotch Marine Boiler.

The main advantage of having the furnace in the flue is, that it does away with the considerable loss of heat by radiation, which always takes place in an external furnace. The difficulty, however, is that the flue, being the hottest portion of the boiler, will have a greater longitudinal expansion than the shell,

and this may tend to loosen the flue at the ends and cause leakage. This difficulty is partly counteracted by making the flue corrugated. Another reason for making the flue corrugated is, that it is exposed to external pressure and therefore requires some extra strengthening. It is a well-known fact, that a cylindrical shell is much better able to withstand a pressure from within than a pressure from without.

The water below the corrugated flue does not get much into circulation, and may be considerably below the temperature in other parts of the boiler, an additional cause of unequal expansion and contraction. Altogether, poor circulation, and as a consequence long time required for raising steam, are the main drawbacks of the Scotch marine boiler. The steaming capacity, however, is large in proportion to the space occupied, and great height is not required in the boiler room. The cost of this boiler is high, compared, for instance, to the ordinary multitubular boiler, but this is offset by no brick setting being necessary. The only fire brick needed is for the bridge and the combustion chamber.

Figure 54 shows only one flue with one furnace. In larger sizes of this boiler there may be two, three or four flues, with one furnace each.

Fire-Box Boiler.

This boiler consists of a cylindrical shell, or barrel, containing a number of tubes, and a rectangular fire-box, as shown in Figure 55. The cylindrical shell is prolonged beyond the rear end of the tubes to form a smoke-box. The combustion gases pass from the fire-box through the tubes to the smoke-box. The fire-box has double walls and a double top, the inside of which is flat, while the outside is the prolongation of the upper part of the cylindrical shell. The other part of the cylindrical shell closes off against the rear wall of the fire-box, to which it is riveted. Near this joint several hand holes are provided for cleaning the space in the double wall. In the head of the smoke-box there is a door, giving easy access for examination of the tubes and removing of soot and dirt.

This is the type of boiler that is used in locomotives, and is therefore often called locomotive boiler. In the form here shown

it is also used for stationary work. The tubes are smaller and spaced much more closely than in the ordinary multitubular boiler, and water circulates in the double walls of the fire-box. Therefore this boiler has a large heating surface in proportion to the floor space it occupies. It is used in small factories, where the space is limited, where it is inconvenient to put in foundation and setting of brick work, and especially where the steam consumption is not continuous, and it is desirable to generate steam very quickly. These boilers are seldom made in sizes over 75 horse-power, and only for low pressure, under which conditions

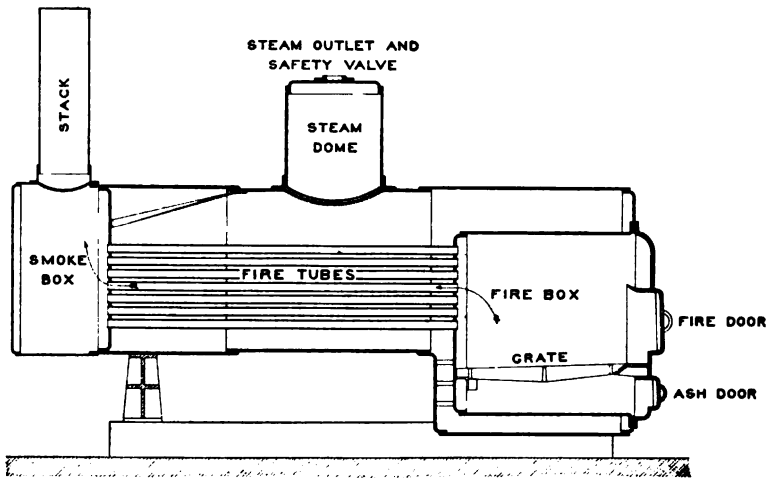


Figure 55. Fire-box Boiler.

they will prove economical and durable. High steam pressures are apt to cause leakage, and in the larger sizes these boilers are more expensive than other types.

Upright Tubular Boiler.

This boiler consists of an upright cylinder, having a concentric flat-topped furnace in its lower end. A common design is shown in Figure 56. Tubes extend from the top of the furnace to the head of the boiler and the gases pass through them into a hood covering the top of the boiler, and thence generally into the chimney, extending upwards from the hood.

This type of boiler is strong, but not very durable, on account of the difficulties involved in examining, cleaning and repairing. It is impossible for a man to get between the tubes or between the shell and tubes. Several hand holes are provided in the shell outside the fire-box, which are the only means of cleaning the boiler. Therefore these boilers can be used with good feed-water only. They require close attention and need

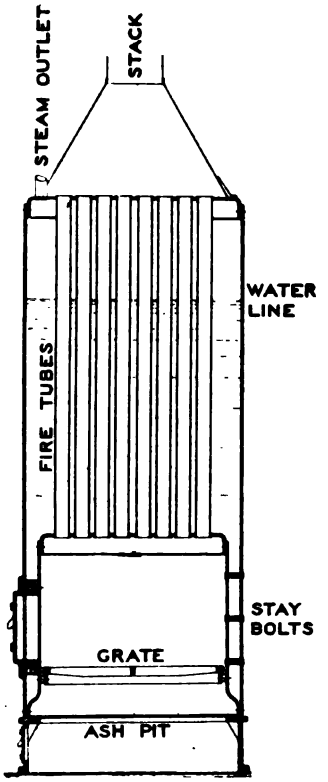


Figure 56. Upright Tubular Boiler.

repairs often. The circulation is very defective and the boiler is wasteful of fuel. This is evident, considering that in the short passage from furnace to stack the combustion gases escape at a high temperature, not having had an opportunity to give off much of their heat to the boiler water. This boiler generates steam quickly, as it has only a small water body.

The top head of the boiler and the upper parts of all the tubes are above the water line and therefore liable to become injured by overheating. This is because steam is a much poorer conductor of heat than water. To avoid this difficulty a *submerged tube sheet* is sometimes put in below the water line, the tubes ending in this. The hood then extends from this tube sheet up to the head of the boiler. As in this construction the hood is exposed to external steam pressure, it must be strong and carefully stayed; therefore this improvement in various ways makes the boiler more expensive.

The small vertical boiler is used when a boiler is wanted that requires very little floor space, is low in price, and gives steam quickly, while durability, fuel economy, liability to fluctuating steam pressure (on account of the small water volume), dry steam, and smokelessness are of secondary consideration. This boiler is often mounted on wheels, and is much used as a portable boiler, more especially in connection with hoisting engines, drilling machinery and pumps.

It should be stated that when a vertical boiler is made high enough, i. e., 20 to 30 feet, most of the short-comings are obviated. Then also the injury to the tube parts above the water line is less, as the combustion gases, passing through the long tubes, have had time to give off much heat and are considerably reduced in temperature.

This type of boiler, as well as those preceding, require no masonry, but should be well protected against loss of heat from radiation by suitable non-conducting covering.

Horizontal Return Tubular Boiler.

Figures 57, 58 and 59 show a longitudinal section and two cross sections of this boiler, which is the most common in this country. It is classified as a horizontal, fire-tube boiler, externally fired. It consists of a cylindrical shell, two flat heads or tube-plates and a number of fire-tubes, 3″ to 4″ in diameter, extending from head to head. The shell extends beyond the front head and forms the smoke box, which is closed with a cast iron head with large doors giving easy access to the tubes. As the cut shows, a brick setting and a cast iron front, with fire doors and ash doors, are required to make the boiler complete and ready

for use. The products of combustion pass back over the bridge, along the lower half of the shell, then forward through the tubes to the uptake and to the chimney. As the tubes in all the boilers so far described are extinguishers of flame, it is highly necessary, in order to insure the greatest efficiency, that provisions be made in the furnace and combustion chamber for the complete combustion of the gases before they enter the tubes.

This type of boiler is now designed to give good water circulation, is easy to clean and repair and has attained a maximum in economy of fuel. It is strong, durable and inexpensive and occupies a moderate floor space, and is decidedly the best stationary boiler for moderate steam pressure and for the small steam user. When properly constructed and managed its liability to explosion is small, but when it does explode the damage is usually large. This boiler is distinctively an American type.

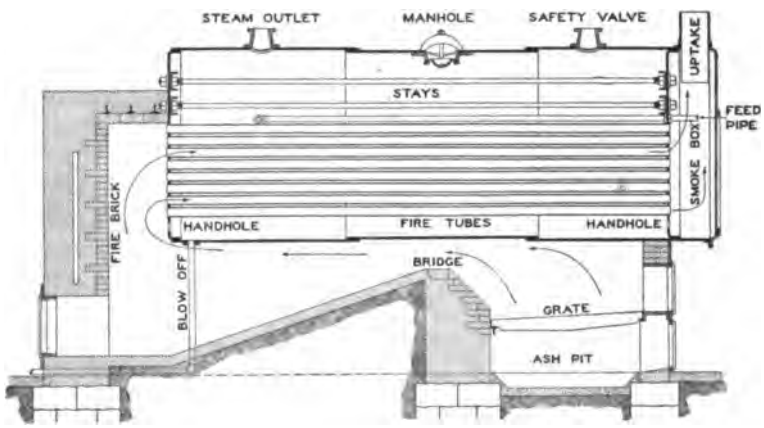


Figure 57. Horizontal Return Tubular Boiler.

The boiler shown in the cuts has two steam nozzles. The main steam pipe should be connected to the nozzle near the rear end of the boiler, as the ebullition is not so violent at that point, and therefore the probability of water being thrown into the steam pipe is less. The nozzle near the front end, which is right above the fire, may conveniently be used for attaching the safety valve. In case this boiler has a dome, steam pipe connection is made to the top of the dome.

The feed pipe, in this type of boiler, should preferably enter

the front head a little below the water line, near the shell, then run towards the rear head, as shown in the cut, and end in a perforated pipe across the boiler (see also "Feeding Arrangement"). The blow-off is usually arranged as shown, and the boiler set with slight pitch toward the rear, so that mud and loose scale will collect near the blow-off pipe.

The boiler is usually supported by two pairs of brackets (lugs), of cast iron or pressed steel, riveted to the shell, the front brackets simply resting in the side walls, while the

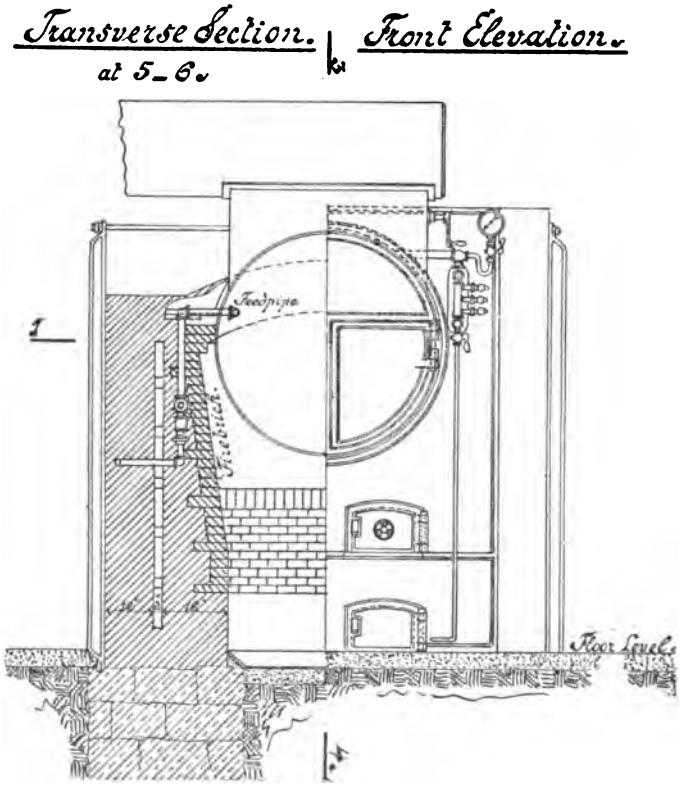


Figure 58. Horizontal Return Tubular Boiler. (Front Elevation.)

rear brackets should rest on rollers moving on an iron plate, or by some other device be allowed a certain freedom of motion to avoid strains, as the shell will expand more than the walls. For this reason there should also be left a small space between the rear tube sheet and the brick work behind the same. The fire

brick lining of the side walls must not be made to receive any of the pressure exerted by the lugs, as it has to be renewed now and then. The boiler may also be suspended from two channel iron girders, resting on two pairs of I-columns built into the brick setting (Figure 60). In this case the shell is provided with two pairs of hanger bars instead of the brackets mentioned above. This method of supporting is the more expensive, but also the best. Brick work and boiler shell may expand and contract independently. Brick work may be removed and repaired without blocking up the boiler and without disconnecting the pipes, and is less liable to settling and cracking, and the piping is not endangered by the boiler settling together with the brick work.

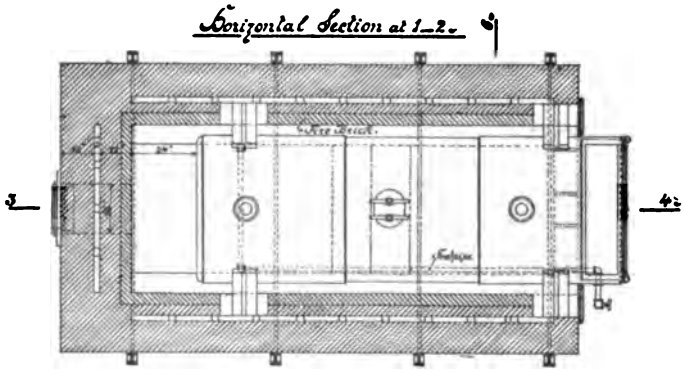


Figure 59. Horizontal Return Tubular Boiler. (Horizontal Section.)

The *boiler-setting* is made of hard, well-burned brick laid in cement or mortar. The side walls are braced by three pairs of buck staves with through rods running across over the tops of the boilers and underneath the floor level (Figure 59). The foundation may be a solid bed of concrete under the whole length and width of the boiler, or under the walls only, as in the cuts, if the soil is good, and then made of concrete or of masonry work. The foundation should be not less than 2 feet deep. The rear and side walls are usually double walls with 2-inch air space between. Between the rear wall and the rear head there should be a space of 24 inches for the larger sizes (above 60″ diameter) down to 16 inches for the smallest boilers of this type. The bridge and all parts of the walls that are

directly exposed to the fire are lined with fire brick. The purpose of the bridge is to direct the flames upwards against the boiler shell, and at the same time support one end of the grate bars, the other end resting on the cast iron front. For boilers of 60″ diameter and over, the space between bridge and shell should be 12 inches down to 8 inches for the smallest ones, and the height from grate to shell 24 to 30 inches, down to 18 inches for the smallest ones. The grate should have a fall of 3 inches from front to rear. In Figure 57 the smoke box overhangs the front setting, while the boiler shown in Figure 60 has the *full flush front.* Manufacturers furnish either style. The latter arrangement requires somewhat more length of foundation and side walls, but is also more convenient for the operation. The

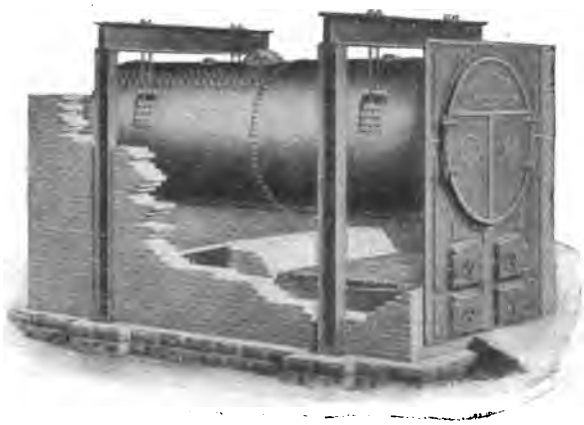


Figure 60. Horizontal Return Tubular Boiler. (Full Flush Front.)

side walls must close in on the sides of the shell in such a way that combustion gases will not come in contact with the plates above the water line. The lower part of the combustion chamber behind the bridge is usually filled with earth and paved with common brick in the way shown in Figure 57; but if wanted, it may be left empty, making the same floor height here as outside the boiler. The rear wall is provided with a door to remove soot and dirt from behind the bridge, and also to facilitate inspection. The top of the boiler must be covered by a brick arch, or by any insulating material on hand, which will stand the high temperature.

Water-Tube Boilers.

This type of boiler is now most generally adopted in large and medium sized power plants, where high steam pressure, comparatively small floor space and safety against explosions are desired, and the higher cost is no obstacle. For convenience, they have been divided into *horizontal* and *vertical.*

Horizontal Water-Tube Boilers.

There are many different makes of this boiler. The common feature, however, is a bank of tubes, 3 to 4 inches in diameter, the water circulating inside, and the combustion gases outside. These banks of tubes are in one way or another connected to drums above, containing some water and besides the proper steam space. These boilers have always comparatively small water space, and therefore fluctuations in pressure are liable to occur, if the fire changes in intensity. On the other hand, the same circumstance makes it possible to obtain steam quickly even with cold feed-water. The difficulty of keeping the large number of joints tight and of cleaning the tubes is greater than in the horizontal tubular boiler. Then again the water-tube boiler has an advantage in that it can be shipped in sections, and the total weight is less. There are very many different makes of this type of boiler, all of which cannot be enumerated here. A brief description, however, is given of the Babcock and Wilcox and of the Heine boiler.

Babcock and Wilcox Boiler.—Figure 61 shows a side view of this boiler (the brick setting is shown in section). The tubes are divided up in a number of vertical sections, each section having at both ends pressed-steel headers of the shape shown in the illustration, into which the tube ends are expanded. Instead of placing the tubes vertically one above the other, they have a staggered arrangement, by which a good spreading of the products of combustion among the tubes is attained. The smaller boiler sizes of this type, up to 170 horse-power, have one, the larger sizes two, horizontal, cylindrical drums on top. Each header has a separate connection to the drum. Opposite each tube end the headers have a handhole, closed with a handhole plate, as shown, the tubes in this way being accessible for cleaning, and a defective tube may be replaced. The headers may either be vertical, as shown here, or at right angles to the tubes.

Baffles of fire tile, fitted in between the tubes, compel the combustion gases to circulate, as shown by the arrows. The circulation of the water is as follows: down from the drum to the rear headers, forward and upward through the tubes, becoming partially vaporized, and then up into the drum where steam and water separate. Water flows continually from the front to the rear of the drum. The rear headers are connected to a mud-drum, from which scale and other sediment may be blown off. The feed-water enters in the front head of the drum, near the bottom.

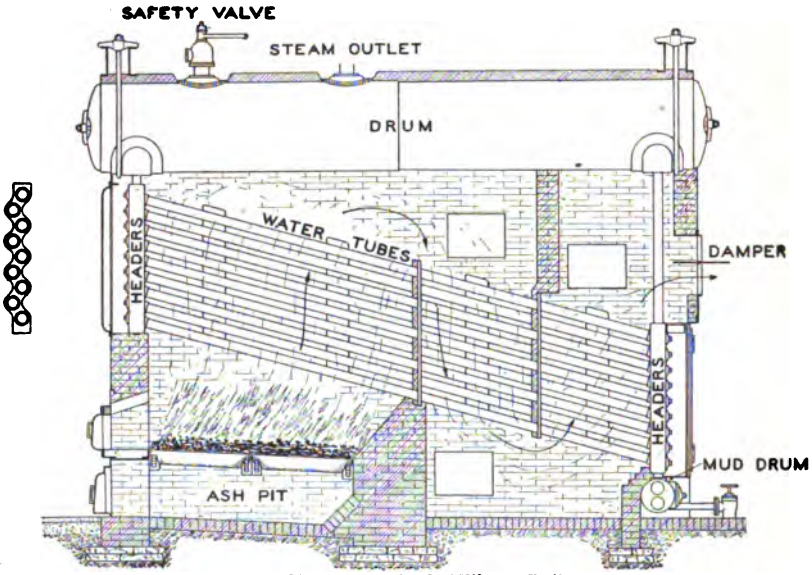


Figure 61. Babcock & Wilcox Boiler.

Cross-girders on columns carry the drums, which in turn carry the tube sections. The brick setting has nothing to carry, but is required as an enclosure for the products of combustion. The space behind the furnace and the outsides of the tubes are cleaned from doors in the side walls; therefore, when a long row of Babcock and Wilcox boilers are installed, they may be built together two and two, but each of these batteries of two boilers must have five feet of clear space on each side. On the other hand, this boiler may be placed directly against the back wall of the boiler house, if necessary. Still, it is highly advisable to

leave a space of at least three feet, if it is possible. In front of the boiler there must be a space as wide as the length of the tubes.

The standard 150 horse-power Babcock and Wilcox boiler has 8 sections with 9 tubes, 4 inches in diameter and 18 feet long, in each, and one drum, 42 inches in diameter and about 20 feet long. The 250 horse-power size has 14 sections, same as above, and two drums, 36 inches in diameter and about 20 feet long.

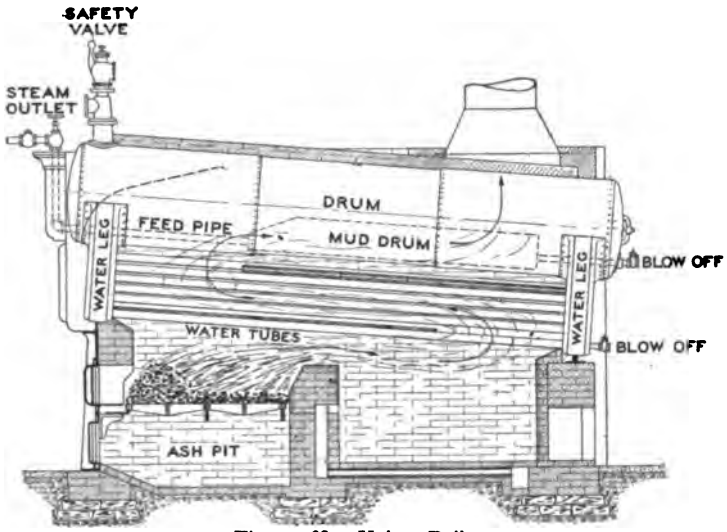


Figure 62. Heine Boiler.

Heine Boiler.—A side view of this boiler is shown in Figure 62 (the brick setting is shown in section). The main difference between this boiler and the Babcock and Wilcox is, that to each end of the drum is connected a box of steel plate, a so-called water-leg, and the tube ends are expanded into the inside plates of these two water-legs. Therefore in this boiler the tubes are not divided into sections. The water-legs must of course be well stayed. Opposite each tube end is a handhole with handhole plate. The Babcock and Wilcox boiler, as in various other makes, has the drums horizontal and the tubes placed on an incline, while in the Heine boiler both drums and tubes are inclined at an angle of about 22 degrees with the horizontal. Nevertheless, all of these boilers are classified as horizontal water-tube boilers.

The water circulation is as described for the Babcock and Wilcox boiler, and also here baffles of fire tile direct the flow of the combustion gases. A special feature of this boiler is the small drum, called the mud drum, inside the main drum. Into this the feed-water is discharged, and, having been heated in the front part of the boiler and thereafter flowing slowly through the mud drum, it may deposit some of its scale-forming substances. Sediment accumulated in this way is blown off through the upper blow-off valve shown in the illustration. A mud drum under the rear end of the tubes, as in the Babcock and Wilcox boiler, is not required, as the rear water-leg serves as such, and therefore the blow-off for loose scale and other solid impurities is connected directly to this. A deflecting steel plate in the front end of the main drum prevents water sprays from being carried into the steam pipe. However, it is advisable, in any make of horizontal water-tube boilers, to pass the outgoing steam through a perforated pipe, also called a *dry pipe,* such as shown in Figure 54.

The front water-leg rests on cast iron columns, built into the brick work; the rear water-leg rests on the brick work, plates and rollers being inserted to allow the boiler to expand independently. The drum is supported by the water-legs. In this boiler, provisions are made for cleaning the space behind the furnace and the outside of the tubes from the rear and from the front of the boiler. Therefore any number of boilers may be set in a row, close together; but sufficient space in the rear is absolutely necessary.

The standard 150 horse-power Heine boiler has 86 tubes, 3½ inches in diameter and 18 feet long, and one drum, 42 inches in diameter and about 21½ feet long. The 250 horse-power size has 138 tubes, same as above, and one drum, 48 inches in diameter and about 21½ feet long. Boilers over 270 horse-power are built with either two smaller drums or one larger.

Vertical Water-Tube Boilers.

The vertical boilers previously mentioned were of the fire-tube type. Vertical boilers with *water-tubes* have also been designed and built.

Wickes Boiler.—This boiler is illustrated by Figure 63.

The top consists of a cylindrical steam and water drum, into which the ends of the vertical tubes are expanded. The bottom is a cylindrical mud drum. The steam drum is arranged to give a height of 66 inches from the water line to the crown sheet, where the steam outlet is located.

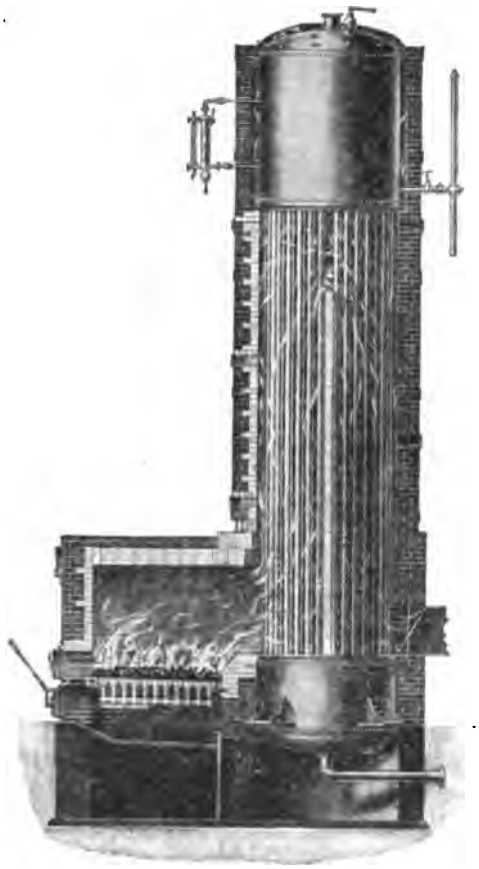


Figure 63. Wickes Boiler.

The tubes are separated into two sections by means of fire-brick tile, the tubes in the part nearest the furnace being called risers, whereas those in the rear are called down-comers, the heated water rising through the front tubes, and the cooler water flowing down through the rear tubes, thus maintaining circula-

tion. The feed-water enters in the upper drum, and the water level is such that all the tubes and upper tube sheet are completely covered. A baffle plate in the top drum deflects the water to the down-comers.

The furnace of this boiler is external, and is built entirely of brick, forming what is called a Dutch oven type.

Stirling Boiler.—A longitudinal section of this boiler is shown in Figure 64. It is a water-tube boiler, but essentially

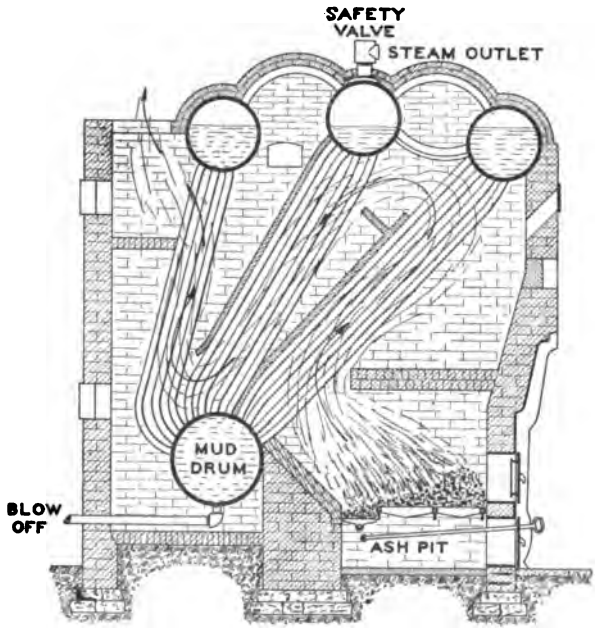


Figure 64. Stirling Boiler.

different from any of the types already described. The larger drum at the bottom is called the mud drum; the three drums at the top have a steam space and also a water space, as indicated in the figure. There are short tube connections, not shown, between the three steam spaces, and a similar connection, below the water line, between the front drum and the middle drum. The feed-water enters the rear drum, flows down to the mud drum through the rear bank of tubes, which mainly acts as a feed-water heater. If the water contains much scale-forming matter, heated in this way it will deposit most of it in the mud drum, whence

it is blown out at intervals. The further circulation of the water is as follows: up through the front bank of tubes, which are the hottest, to the front drum, over to the middle drum through the tube connection mentioned above, and down again to the mud drum through the center bank of tubes. Baffles of fire tile and the fire brick arch over the grates direct the flow of the combustion gases, as shown. The drums are accessible for cleaning through manholes provided in the heads.

This boiler is suspended in a way similar to the Babcock and Wilcox boiler. The cleaning has to be done from the side,

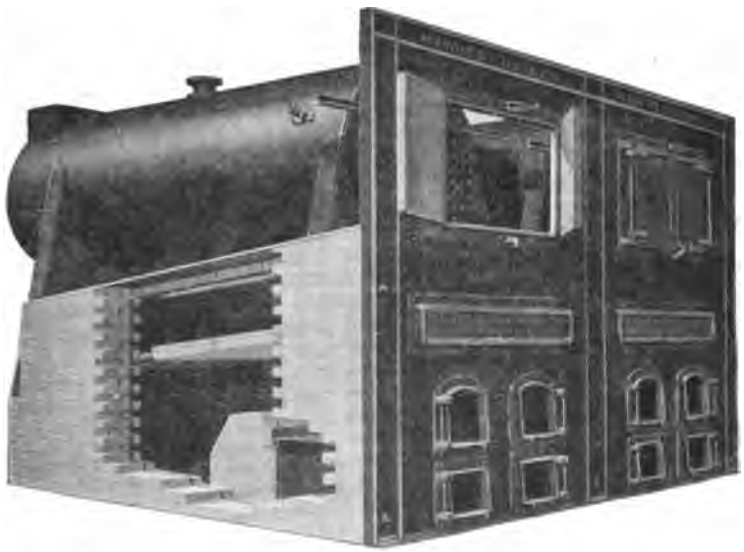


Figure 65. Hawkes Boiler.

so only two of these boilers can be set close together in one battery. Where the feed-water is hard, this boiler has an advantage over most other types, not only on account of the precipitation of scale-forming substances in the mud drum, but also because the very rapid water circulation through nearly vertical tubes offers a condition not favorable to scale formation. The construction gives the combustion gases a good chance to give off much heat before entering the chimney.

The Stirling boiler is easily adapted to different conditions of floor space, height, etc. For instance, a 250 horse-power

boiler with setting can be built 16 feet wide, 14 feet long and 12 feet high, or 9 feet wide, 19 feet long and 24½ feet high, or in other variations, with over-all dimensions between the limits here given.

Combined Water-Tube and Fire-Tube Boilers.

The *Hawkes boiler,* shown in Figure 65, is a combination of the return tubular boiler and the water tube type. In this construction, the shell plates, seams, joints, hand hole plates and connections do not come into direct contact with the furnace gases. The heating surface is all tube surface.

The gases travel first under the lower water tubes, then under the upper water tubes, and finally through the fire tubes in the upper shell. The circulation is rapid, the scale and mud deposits being carried to the bottom of the upper shell away from fire action.

The setting requires little brickwork, and has a steel casing, whereby air leakages are reduced to a minimum. The furnace is a reverberatory type and is a part of the boiler.

AUXILIARY STEAM BOILER APPLIANCES.

Feeding Arrangement.

The question of introducing the *feed-water* into the boiler in the proper way is of much importance, and depends to some extent on the type of the boiler, the temperature in the boiler, the temperature of the feed-water, etc. Even for the same boiler under the same conditions, experienced engineers may differ in their opinions as to the best solution of this question. One point that must be observed is, that the full stream of feed-water should not be directed against a highly heated boiler plate. As the feed-water is always colder than the water in the boiler, this condition will cause local contraction and eventually a leak. The colder the feed-water, the hotter the boiler water, and the more irregular the feed-water supply, the more detrimental will be the effect of this.

The arrangement shown in Figure 57 is recommended for horizontal return tubular boilers. The feed-water here becomes well heated by its passage through the boiler water. The same

object is attained by spraying the feed-water into the steam space of the boiler, by the arrangement shown in Figure 66, or by letting it overflow from an open trough, or by connecting the feed pipe to a piece of perforated pipe. In these ways the feed-water is well heated before mixing with the boiler water, the air is removed from it, and certain scale-forming substances are precipitated. There is the disadvantage that the steam pressure may be affected when the feed-water supply is suddenly increased, especially when it is comparatively cold, and, furthermore, it must not be overlooked that when the proper precautions are taken, there is, in many cases, an advantage in delivering the feed-water in a full stream into the boiler water, thereby creating currents helpful to the water circulation.

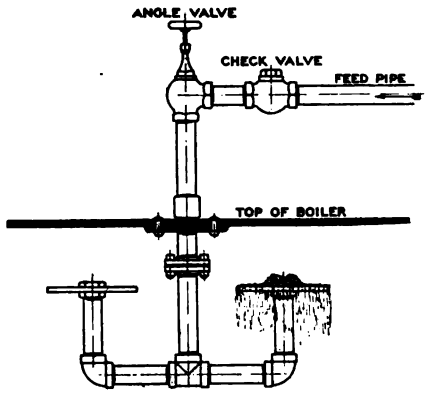


Figure 66. Boiler Top-Feed.

The method often used, of introducing the feed-water through the blow-off opening, is simple and convenient for the piping contractor, but not recommendable. The more or less cold feed-water is thereby delivered into the coldest part of the boiler, which tends to create a space of dead water and interferes with the proper circulation.

Speaking in general, it may be said that the best practice is to allow the feed-water to enter a few inches underneath the water line, and to continue the feed pipe a fairly long distance inside the boiler.

The feed-water is usually forced into the boiler by a *steam pump* and in some cases by an *injector*. In small boiler plants one steam pump and one injector are generally used and in larger

plants two steam pumps should be installed. Portable boilers are equipped with an injector only, in which case the injector becomes very useful and efficient, because the cold feed-water is heated by the steam which forces the water into the boiler. Thus there is no loss in feeding a boiler with cold water by an injector, provided there is no other and cheaper method of heating the feed-water. However, if the feed-water can be heated with heat that is otherwise wasted, then the injector becomes a very expensive apparatus, and a steam pump is far superior, as it can deliver the water much hotter into the boiler, and the feeding of the boiler can also be done more uniformly. If conditions allow it, uniform feeding and uniform water level in the boiler contribute to fuel economy and to long life of the boiler. The feed-water is generally heated by exhaust steam in a separate vessel called the *feed-water heater,* and where no exhaust steam is available the feed-water is sometimes heated by the waste heat of the chimney gases through an apparatus called an *economizer.* The hotter the water is delivered into the boiler the larger will be its capacity and fuel economy, provided that waste heat is employed for this purpose. The feed-water pipe is usually, or should be, provided with a stop valve and a check valve, both located close to the boiler, the stop valve being between the boiler and the check valve, so that the latter may be examined and, if necessary, repaired or renewed without shutting down the boiler. The check valve serves the purpose of preventing the water in the boiler from backing out. It is not absolutely necessary if there is only one boiler, but even in this case it is better to have it. If a battery of boilers is fed through one feed pipe, it is absolutely necessary to have a check valve on each boiler to prevent the water from backing from one boiler into another.

Blow-Off.

The back end of each boiler should be set one inch lower than the front end so that the boiler can be properly drained. The bottom of the rear end of the boiler is provided with a *blow-off pipe* which should lead into an iron catch basin to the sewer. From the top of the catch basin a vapor pipe leads through the boiler room roof to allow the steam to escape into the air, and only the water to flow into the sewer. The blow-off

pipe should be provided with a cock or a valve made for this purpose. A small piece of scale may lodge on the seat of a valve, making it impossible to close it tightly, but it may not be noticed and will then cause leakage from the boiler. In this respect a cock is better, for when the plug is in the closing position, it is closed; on the other hand, a cock is liable to stick, if it is not operated often enough. Frequently both cock and valve are inserted. The part of the blow-off pipe that is exposed to the fire should be covered with asbestos or fire brick. Boilers should not be emptied as long as steam pressure is on and the brick work is hot, as there is a tendency to bake the sediment on to the metal surface, from which it is only with difficulty removed. A certain portion of the water usually one-half to one gauge (the distance between two gauge cocks on the water column) should be blown off at regular intervals of two to eight hours, according to the nature of the water, when the boiler is in operation, whereby the boiler is relieved of a great deal of scale-forming sediment.

Tubular boilers should be cleaned every two to four weeks, this also depending on the nature of the water. If the scale-forming ingredients of the water and the amount of water fed into the boiler are known, the *sediment* in the boiler can be calculated for the time the boiler is in operation. Supposing the water to contain 15 grains of solid matter per gallon and the boiler to evaporate 3,000 pounds of water per hour, then the amount of sediment deposited in the boiler in 28 days, assuming a gallon of water to weigh 8 pounds, will be

$$\frac{3{,}000 \times 24 \times 28 \times 15}{8 \times 7{,}000} = 540 \text{ pounds.}$$

Steam Gauge.

It is absolutely necessary that every boiler be provided with a trustworthy steam gauge for indicating the pressure in the boiler at all times. In order to avoid fluctuations in pressure, this gauge should be connected directly to the boiler and not to the steam pipe. Gauges should preferably be connected to the water column, as shown in Figure 67, and in the pipe connection there must be a vertical U, or also a coil pipe, which will fill up with condensed water. Thereby the gauge is acted on by water

under the pressure of the steam instead of the steam itself, the high temperature of which would do harm to the gauge.

Water-Column.

It is fully as important to know the height of the water in a boiler as to know the steam pressure. In order that the water-level in a boiler may be ascertained at any time, a water column, on which are mounted one *gauge-glass* and three *gauge cocks,* is used. As long as both ends of the gauge-glass are in unobstructed communication with the boiler, the gauge-glass will, of course, show the water-level actually existing in the boiler.

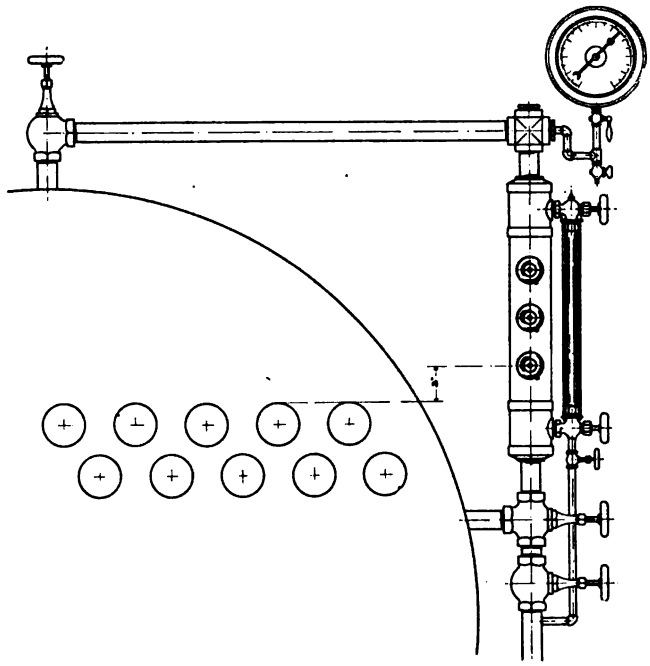


Figure 67. Boiler Gauge Glass.

Gauge-cocks are used to verify the water-level shown by the glass, and are used, in case the latter is broken, during the time a new glass is inserted. These cocks are usually placed three inches apart, the upper showing the highest, the middle the normal and the lower the lowest allowable water-level in the boiler. Figure 67 shows a *water-column* properly connected to a boiler.

The column and glass are provided separately with blow-off connections, which should be used frequently and at regular intervals. A water-column provided with high and low water alarms (not shown in the Figure) should, if possible, be used. Such a column by action of a float attached to a whistle, automatically blows the latter when the water reaches the high or low safety level, thereby warning the engineer in charge, that the water-level in the boiler is either too high or too low.

Fusible Plugs.

In some states and cities the insertion of fusible plugs at the lowest water line in boilers is compelled by law. All boilers under the jurisdiction of the U. S. Marine must be provided with fusible plugs of Banca tin, at least one-half inch in diameter. A fusible plug is made of brass, with a core of some alloy that will melt when the plate in which it is inserted becomes unduly over-heated. The plug should be inserted in shell, head or tube under the normal water line at a point below which it is objectionable, or even dangerous, to let the water level sink during the operation of the boiler. As long as the normal water level is kept up, the heat is carried away by the water, by conduction, comparatively fast. However, when only a thin layer of water is left, or none at all, the plug and the part of the boiler nearest to it will become hotter, as the steam is a poor conductor of heat. The fusible alloy then melts out of the plug, and the escaping steam gives warning to the fireman. Newton's fusible alloy consists of about 5 parts of bismuth, 3 parts of lead and 2 parts of tin; it melts at 202° F. There are many different alloy mixtures of metal which possess different melting points. The certainty of action of a fusible plug depends on proper exposure, that is, scale should not be allowed to accumulate on the face of the plug.

Safety-Valves.

Safety-valves must be placed on the highest spot of the boiler and not be separated therefrom by stop valves. A safety-valve serves the purpose of automatically relieving the boiler of a dangerously high pressure, and of giving notice that something is wrong. There are two classes of safety-valves, the lever safety-valve (Figure 68) and the spring or pop safety-valve

(Figure 69). The *lever-valve* can be given a larger or smaller resistance by sliding the weight along the lever arm either from or towards the valve, as the case may require. The *pop-valve* is more reliable and is therefore mostly used. It is equipped with a screw by which the tension of the spring can, within certain limits, either be increased or decreased. High-pressure boilers should be equipped with two safety-valves, one loaded slightly heavier than the other. This has been found to give the best satisfaction, and so-called twin valves (two pop-valves mounted in one casting) are generally used for this purpose.

It should be borne in mind that a safety-valve is not a preventive of a boiler explosion, as it only releases that pressure which is in excess of its load, thus if a safety-valve is loaded to 150 pounds pressure and the boiler explodes with 125 pounds, then the fault lies in the loading of the valve and not in its proper function. Safety-valves should not be loaded higher than one-fifth of the bursting pressure of the boiler. They should be tried daily by lifting them off their seat.

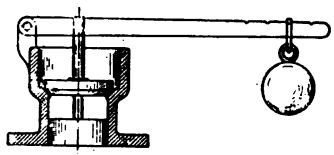


Figure 68. Lever Safety Valve.

Dimensions of Safety-Valves.—A great number of rules are to be found in city ordinances, state laws, text books, etc., concerning the proper area of a safety-valve. They are so conflicting, however, that by applying them on the boiler in the example below, we obtain results varying from 6.5 to 48 square inches.

The Rule of the U. S. Supervising Inspectors of Steam-vessels (as amended in 1909) is as follows:

"The area of all safety-valves shall be determined in accordance with the following formula:

$$a = 0.2074 \times \frac{W}{P}$$

a = area of safety-valve, in square inches, per square foot of grate surface.

W = pounds of water evaporated per square foot of grate surface per hour.

P = absolute pressure in pounds per square inch.

"The value of *a* multiplied by the square feet of grate surface gives the area of safety-valve or valves required. When this calculation results in an odd size of safety-valve, use the next larger standard size."

Example.—Boiler-pressure, P = 100 lbs. gauge = 115 absolute. Grate surface = 40 sq. ft. Water evaporated per pound of coal = 9 lbs. Coal burned per sq. ft. of grate surface per hour = 10 lbs. Consequently, evaporation per square foot of grate surface per hour, W = 10 × 9 = 90 lbs. What should be the diameter of the safety valve?

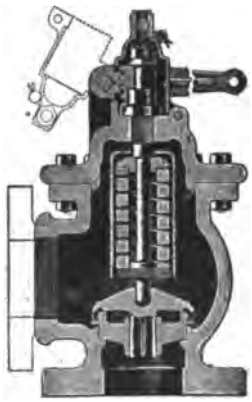


Figure 69. Spring Safety Valve.

Solution.—$a = 0.2074 \times \frac{90}{115} = 0.162$ square inch per square foot of grate surface.

Therefore area of safety-valve = 40 × 0.162 = 6.5 sq. in.

Diameter = $\sqrt{\frac{6.5}{0.7854}}$ = approximately 3 inches.

"Each spring-loaded valve shall be supplied with a lever that will raise the valve from its seat a distance of not less than that equal to one-eighth of the diameter of the valve opening.

"Whenever the area of a safety-valve, as found by the rule, will be greater than that corresponding to 6 inches in diameter,

two or more safety-valves, whose combined area shall be equal at least to the area required, must be used.

"The seats of all safety-valves shall have an angle of inclination of 45 degrees to the center lines of their axes."

The lever mentioned above, with which spring-loaded valves should be supplied, has nothing to do with the automatic action of the safety-valve. It is intended for occasional lifting of the valve from its seat by the fireman, which should be done to insure that it is in good condition. The lift of one-eighth of the diameter, mentioned in this connection, is also much more than the valve is lifted, when blowing off automatically on account of too high pressure. Then the lift seldom exceeds one-tenth of an inch, and is in many cases only half of that.

For lever safety-valves the *distance at which the ball must be placed* from the fulcrum for a required pressure can be found according to the following formula:

$$p \times a \times d = W \times D + w \times d.$$

p = pressure in pounds per square inch at which valve will lift.
a = area of valve in square inches.
d = distance in inches of valve stem from fulcrum.
W = weight of ball in pounds.
D = distance in inches of ball from fulcrum.
w = effective weight of lever, valve and stem in pounds.

$$\text{Then } p = \frac{W \times D + w \times d}{a \times d}$$

$$D = \frac{(a \times p - w) \times d}{W}$$

$$W = \frac{(a \times p - w) \times d}{D}$$

Example.—At what pressure will a 4-inch safety-valve blow off with the valve stem 3.5 inches from the fulcrum, and having a lever, valve and stem of an effective weight of 25 pounds, and a 50-pound ball 26 inches from the fulcrum? (See Figure 68).

Solution.—$p = \frac{50 \times 26 + 25 \times 3.5}{4 \times 3.5} = 99$ pounds.

Example.—At what distance would the ball have to be placed from the fulcrum, if it weighed 75 pounds and if the steam pres-

sure at which the valve should blow off were 100 pounds, other conditions remaining the same as in the previous example?

Solution.—$D = \frac{(4 \times 100 - 25) \times 3.5}{75} = 17.5$ inches.

Steam-Domes, Drums and Dry-Pipes.

Steam-domes or drums are expensive adjuncts to horizontal return tubular boilers and are useless on boilers which are well designed and proportioned with a view of dispensing with either. They are attached to boilers under the mistaken idea of increasing the steam space and procuring dry steam. They, however, effect nothing in the way of drying steam, but act, on the contrary, as a condenser, and the extra amount of steam space they afford is of very little value. A properly designed dry-pipe (see Figure 54) is less expensive and much more effective as far as the drying of the steam is concerned. A dry-pipe has perforations on the upper side, and may run nearly the whole length inside the boiler. As a rule, drums and domes do more harm than good. If not perfectly constructed, steam-domes are an element of weakness to the boiler.

Mud-Drums.

Mud-drums are made of varying sizes from 16 inches up to 30 inches in diameter, and are usually placed cross-wise under the rear end of a boiler or sometimes under a battery of boilers, with a short upright connection of large diameter to each boiler. They are often arched over with brick to prevent direct contact with the flame and heat. The main object of a mud-drum is to serve as a receptacle for mud and dirt and to bring the feed-water temperature approximately to that prevailing in the boiler. Upon examination, however, the upper part of the drum will be found much warmer than the bottom, and the result, as might be expected, is unequal expansion and contraction, producing leaks in the ring-seams and neck-flanges, as well as corrosion at those places, together with grooving of the boiler, and internal corrosion of the sheets of the drum. If a mud-drum is used at all, the feed-water should not enter it, but should be fed into the boiler from the top. It should preferably be made of cast iron, and will then resist corrosion and most of the above-named troubles satisfactorily.

Feed-Water Heaters.

Feed-water heaters are apparatus in which the heat of the exhaust steam is employed for raising the temperature of the feed-water as near to the boiling point as possible. This heat, which would otherwise be wasted, is thus utilized and fuel is saved. If no exhaust steam is available, the heat from the chimney gases may be utilized for this purpose, and even if live steam

Figure 70. Closed Heater.

is the only heating medium on hand, it pays to spend some of it on the heater, and not allow the feed-water to enter the boiler too cold.

Besides heating the feed-water, the heater often performs the function of purifying it to a certain extent, provided the heater is of sufficient size and properly constructed.

Closed Heaters.—Closed heaters (Figure 70) are usually vertical cylindrical vessels, filled with brass tubes, and are con-

structed like steam condensers. The water either passes around the tubes and the exhaust steam through them, or the steam passes around the tubes and the water through them. The heater is generally placed between the boiler and boiler feed pump, thereby being under the same pressure as the boiler.

Ordinarily the saving of fuel is about 1% for every 11° F. added to the feed-water temperature by the heater. These heaters are frequently overrated by the manufacturers, this being the principal reason that so many heaters fail to perform the duty

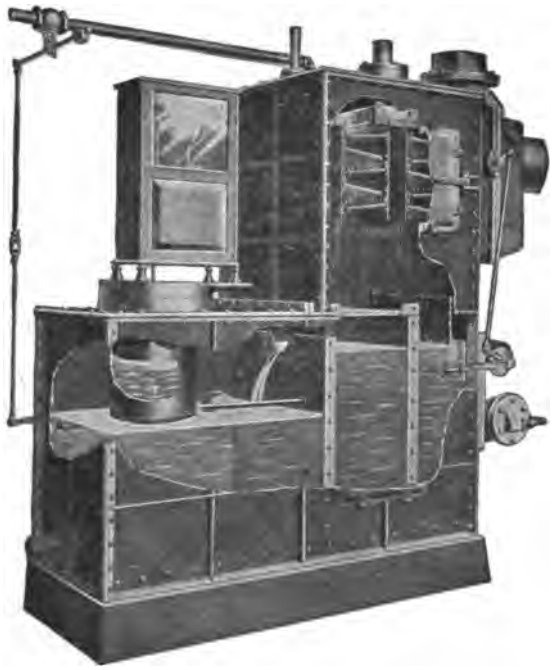


Figure 71. Cochrane Feed-water Heater.

for which they are bought. They usually contain one-third square foot of tube surface per horse-power, but this has been proven in practice to be insufficient, and heaters of this style should contain from three-fifths to one square foot of tube surface for each boiler horse-power.

Open Heaters.—The so-called open heaters (Figure 71) are heaters in which the water comes in direct contact with the exhaust steam. The water is brought into spray form in the

heater, the spraying devices being constructed in various manners. It is evident that a closer exchange of heat is possible with this style of heater than with the closed heater. The proper height of water in the heater is regulated by a float. The pump draws the water from the heater and forces it into the boiler. A filter bed, consisting of coke, excelsior, hay or other suitable material, is generally placed at the bottom of the heater.

Advantages of Open Heaters.—For many plants the open heater possesses advantages over a closed heater, without corresponding disadvantages. In the closed heater, the condensation of the exhaust steam is usually lost, while in the open heater it is mixed with the feed-water and is fed into the boiler. In this manner a considerable amount of water free from scale-forming material is also obtained. It has been claimed that this water is unfit for boiler use on account of its containing oil or grease, absorbed from the exhaust steam. This objection, however, does not hold good, as all open heaters are provided with oil extractors and filters, and consequently furnish practically clean water to the boilers.

The most important advantage the open heater has over the closed heater is the purification of the water, which does not take place in the closed heater. If there is a precipitation of impurities in the closed heater it is generally objectionable on account of the difficulty entailed in removing it. The efficiency and capacity are generally lessened, if scale from the water forms a coating on one side, and oil and grease form a coating on the other side of the heating surface, because the heat transmitting power of the metal is greatly reduced. In the open heater there is no metal surface interfering with the heat transmission, since the water mixes directly with the exhaust steam. The water is thereby heated almost to the temperature of the exhaust steam, and passes downward to the settling chamber and through a filter bed before entering the pumps. All impurities that are precipitated between 175° and 210° F. remain in the filter bed. The impurities can be removed at some convenient time. Open heaters have the further advantage of being easily accessible, which is of importance in case cleaning or repairing becomes necessary.

It has sometimes been claimed that the pumping of hot water involves difficulties, but if the proper pump and valves are selected

and the pump is placed so that the hot water flows into it by gravity, there will be no trouble. Pump manufacturers guarantee their pumps for whatever duty they are sold to perform.

Calculation of Saving by Heating Feed-Water.—For finding the saving of fuel which may be expected by raising the temperature of the feed-water a certain number of degrees, the following formula may be used:

$$M = \frac{(t_1 - t) \times 100}{H - h}$$

M = saving of heat in per cent.

H = total heat above 32° F. in the steam in B. T. U.

h = heat in feed-water above 32° F. in B. T. U. before heating.

t_1 = temperature in degrees F. of the feed-water entering the boiler.

t = temperature in degrees F. of the feed-water entering the heater.

Percentage of Saving in Fuel by Heating Feed-Water by Waste Steam per Degree Increase in Temperature.

Initial Temperature of Feed-Water Degrees F.	Gauge Pressure in Boiler in lbs. per square inch.											Initial Temperature of Feed-Water Degrees F.
	0	20	40	60	80	100	120	140	160	180	200	
35	.0872	.0860	.0853	.0849	.0846	.0843	.0841	.0840	.0838	.0837	.0836	35
40	.0875	.0863	.0857	.0853	.0851	.0847	.0845	.0843	.0842	.0841	.0839	40
50	.0883	.0871	.0864	.0860	.0857	.0854	.0852	.0850	.0849	.0848	.0847	50
60	.0891	.0878	.0872	.0867	.0866	.0861	.0859	.0858	.0856	.0855	.0854	60
70	.0899	.0887	.0879	.0875	.0873	.0869	.0868	.0865	.0864	.0862	.0861	70
80	.0907	.0894	.0887	.0883	.0881	.0877	.0874	.0873	.0871	.0870	.0869	80
90	.0915	.0902	.0895	.0890	.0889	.0884	.0882	.0880	.0879	.0877	.0876	90
100	.0924	.0910	.0903	.0898	.0897	.0892	.0890	.0888	.0887	.0885	.0884	100
110	.0932	.0919	.0911	.0907	.0905	.0900	.0898	.0896	.0895	.0893	.0892	110
120	.0941	.0927	.0920	.0915	.0913	.0908	.0906	.0904	.0903	.0901	.0900	120
130	.0950	.0936	.0928	.0923	.0921	.0917	.0914	.0912	.0911	.0909	.0908	130
140	.0959	.0945	.0937	.0932	.0930	.0925	.0923	.0921	.0919	.0918	.0916	140
150	.0969	.0954	.0946	.0941	.0939	.0934	.0931	.0929	.0928	.0926	.0925	150
160	.0978	.0963	.0955	.0950	.0948	.0943	.0941	.0938	.0936	.0935	.0934	160
170	.0988	.0972	.0964	.0959	.0959	.0952	.0949	.0947	.0945	.0944	.0942	170
180	.0998	.0982	.0973	.0968	.0966	.0961	.0958	.0956	.0954	.0953	.0951	180
190	.1008	.0992	.0983	.0978	.0975	.0970	.0967	.0965	.0963	.0962	.0960	190
200	.1018	.1001	.0993	.0987	.0985	.0980	.0977	.0975	.0973	.0971	.0970	200

Example.—What saving of heat is accomplished by heating the feed-water from 60° F. to 204° F., if the steam gauge of the boiler indicates 100 pounds pressure?

Solution.—From the steam table we find that the total heat above 32° F. in steam of 100 pounds pressure is 1188.8 B. T. U., and by substituting the figures of the example in the above formula we obtain:

$$\frac{(204-60)\times 100}{1188.8-28} = 12.4\% \text{ saving of heat.}$$

The table on page 205 gives the percentage of saving in fuel by heating the feed-water by waste steam, per degree increase in temperature. The use of the table may be made clear by an example, for comparison using the same example as above.

Solution.—Locating the point which corresponds to 60° F. temperature and 100 pounds pressure, it is found to be 0.0861 This is the per cent saving for 1 degree increase in temperature. For an increase of $204-60=144$° F. in temperature, the saving is 144×0.0861, or approximately 12.4%.

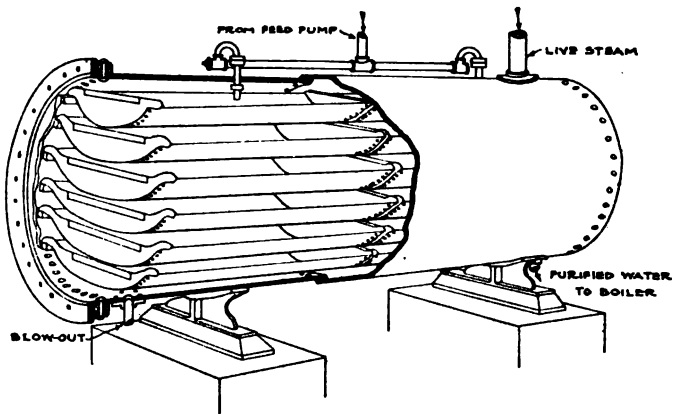


Figure 72. Live Steam Purifier.

Live Steam Purifiers.

A device of this kind is shown in Figure 72. It consists of a cylindrical vessel made of heavy boiler steel, with a large manhole in one end and a number of oval-shaped pans inside, which extend in a horizontal position nearly the entire length of the vessel. The pans are arranged in vertical rows, the water being

fed into the top pan, from which it overflows into the next lower one, and so on until the water reaches the bottom of the vessel, from which it flows by gravity into the boilers.

These purifiers are always placed above the boilers. The vessel is in direct connection with the boilers and consequently under the same steam pressure and temperature. Most of the scale-forming ingredients are deposited in the pans on account of the water being subjected to the same high temperature as the steam, before it enters the boilers. The pans are removed when necessary, usually when the boilers are washed, and the scale and slush taken out. Cold water is first run through, causing the pans to contract suddenly, thereby cracking off much of the hard scale. The pans should be cleaned immediately after being taken out; if allowed to stay exposed to the air with the scale on, some soft scale may harden and become less easily removable.

Live steam purifiers are mostly used where the water contains large quantities of sulphates, as these ingredients will not precipitate at the ordinary temperatures at which an exhaust steam heater heats the feed-water.

Feed-Water Economizers.

A feed-water economizer is a device consisting of a series of tubes arranged vertically, and placed in a chamber constructed in the main flue between the boiler and the chimney. There must, however, be a by-pass flue and suitable dampers, so that it is possible to repair the economizer and still keep the boilers running. The tubes are usually 4 inches in diameter and about 6 to 10 feet long. Scrapers are fitted to these tubes and arranged to be driven in any convenient manner, so that they are continuously in action, removing the soot as fast as it is deposited on the tubes.

There is a rapid loss of efficiency if the tubes are allowed to become dirty, which may occur externally if soot is allowed to collect, or internally if mud accumulates in the tubes, or if scale is formed. As a check on the efficiency, thermometers should be inserted, so that the temperature of the water will be indicated on entering as well as on leaving the economizer. The difference between the temperatures recorded becomes less as

the efficiency falls from any cause, and an examination should be made when this difference is observed to become smaller.

An economizer is usually installed where no exhaust steam is available, i. e., where the exhaust steam from the engines is expanded to lower pressure than that of the atmosphere. Economizers are expensive and should not be installed in small plants nor in places where the temperature of the flue gases upon leaving the boilers does not exceed 400° F. The usual velocity of the water through the economizer is about one-half inch per minute and the required tube area should be calculated accordingly. For instance, if a boiler evaporates 3,000 pounds of water per hour the required number of 4-inch tubes would be:

$$\frac{3{,}000 \times 144 \times 24}{60 \times 60 \times 12.57} = 229 \text{ tubes.}$$

The working of this example is as follows: The amount of water evaporated per minute is $\frac{3{,}000}{60} = 50$ pounds, or assuming 1 cubic foot of water weighs 60 pounds, $\frac{50}{60} = \frac{5}{6}$ cubic foot. The pipe area required to allow one cubic foot of water to pass through at a velocity of 12 inches per minute would be $\frac{12 \times 12 \times 12}{12} = 144$ square inches, and for a velocity of one-half inch per minute, $12 \div \frac{1}{2} = 24$ times as much, or $144 \times 24 = 3{,}456$ square inches. Hence 5/6 cubic foot requires $3{,}456 \times 5/6 = 2{,}880$ square inches, and this divided by the area of a 4-inch diameter pipe, i. e., $4 \times 4 \times 0.7854 = 12.5664$, or 12.57, amounts to $\frac{2{,}880}{12.57} = 229$ tubes.

If the tubes were 10 feet long, the water would be exposed to the heat for $\frac{10}{2.5} = 4$ hours, since the water travels one-half inch per minute, which equals 30 inches or 2.5 feet per hour.

Determination of Temperature of Combustion and Flue Gases.

The temperature in the combustion chamber can be measured by a so-called *pyrometer,* for instance, *the platinum pyrometer.* The temperatures of the escaping flue gases, however, should not be ascertained by pyrometers, because these instru-

ments are not absolutely reliable. For this purpose, the *mercury thermometer* containing compressed *nitrogen* in the tube above the mercury, should be used. These thermometers may be used at considerably higher temperatures than ordinary mercury thermometers. If no such instrument is at hand, the temperature of the flue gases can be determined approximately by the following method:

A piece of iron, the weight of which has been accurately determined, is inserted into the flue, and is left there until it can be taken for granted that it is heated to the temperature of the flue gases. It is then quickly removed and at once submerged in water, the weight and temperature of which must previously have been determined with accuracy. The temperature of the water will rise, and the final temperature is determined by means of an accurate thermometer. From the increase in temperature and the other data known, the temperature of the flue gases can be calculated as follows:

$$T = \frac{W \times (t_1 - t)}{w \times s} + t_1$$

T = temperature of the flue gases in degrees F.
W = weight of water in pounds.
w = weight of iron in pounds.
t = initial temperature of water in degrees F.
t_1 = final temperature of water in degrees F.
s = specific heat of iron.

Example.—In applying the above method, the piece of iron weighed 10 pounds, the water 100 pounds, the temperature of the water before submersion of the iron was 50° F., after submersion, 55° F. Specific heat of the iron, 0.1138. What is then the temperature T of the flue gases?

Solution.—$T = \frac{100 \times (55 - 50)}{10 \times 0.1138} + 55 = 494°$ F.

Recording Thermometer.—The recording thermometer is a tube generally filled with ether, one end of which is placed in the substance or liquid of which the temperature is desired, and the other end connected to the gauge. On heating, the ether in the tube expands and presses on the gauge, the latter being supplied with an indicator dial.

Recording thermometers are very valuable instruments, in as much as they record the temperature at all times, and in this manner variations in temperature can readily be detected.

Every boiler room should be supplied with a recording flue gas thermometer, such an instrument indicating at once to the fireman when he fires incorrectly. It shows the folly of charging the furnace with an excessive amount of fuel at one time, through which long periods elapse before re-charging. Furthermore, by having a recording thermometer, the chief engineer is furnished with a record of how the firing has been done.

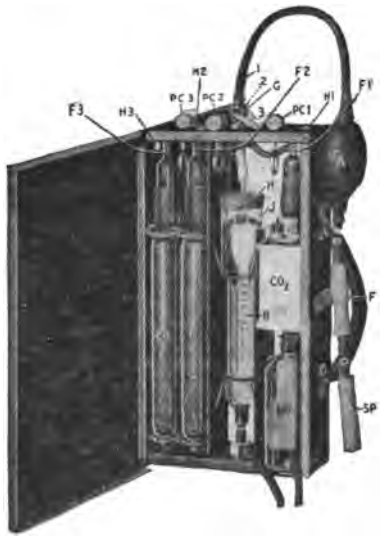


Figure 73. Hays Flue Gas Analyzer.

Flue-Gas Analysis.

The analysis of flue gases is an especially valuable method of determining the relative value of different methods of firing, or of different kinds of furnaces. In making these analyses, great care should be taken to procure average samples, since the composition is apt to vary at different points of the flue. The composition is also apt to vary from minute to minute, and for this reason the drawing of gas should last a considerable period of time.*

* Am. Soc. of Mech. Eng. "Rules for Conducting Boiler Trials, Code of 1899."

Various instruments for the analysis of flue-gas are on the market. They show either the carbon dioxide (CO_2) only, or carbon dioxide, carbon monoxide and oxygen, the latter allowing the calculation of the free air. For determining the three gases, solutions of caustic potash, cuprous chloride and pyrogallic acid, respectively, are used, as they absorb these gases. By the loss in volume caused by the absorption, the percentages may be read off on an indicator. The Hays gas analysis instrument is shown in Figure 73.

The instruments are extremely valuable where analyses are made continuously and charts obtained. Only when this is done can camparisons be made from day to day.

STRENGTH OF BOILERS.

The strength of boilers depends mainly upon the tensile strength of the material of which the boiler shell is made; upon the diameter of the shell, upon the kind of riveting applied, i. e., upon the per cent of strength the joint possesses in comparison to the solid plate, and upon the bracing of the flat surfaces. For externally fired boilers, the plate should not be thicker than one-half inch. The circular seams are usually single-riveted and the longitudinal seams double or triple-riveted. For low pressure, double-riveted lap joints, and for high pressure, butt-joints are used. The latter are usually either double, triple, or quadruple-riveted (see "Rivets," page 213).

The tensile strength of a boiler plate is usually given by manufacturers in pounds per square inch of cross-section. Tensile strength means the pull in pounds it would require to pull apart a bar one square inch in cross-section. The material generally used varies between 55,000 and 65,000 pounds tensile strength. The factor of safety allowed should not be less than 5, which means that the safe working pressure allowed on a steam boiler should not exceed one-fifth of its bursting pressure.

Calculation of Safe Working Pressure.

The usual formula for the calculation of the strength of cylindrical vessels (steam boilers) adopted by the Board of Trade is:

$$P = \frac{t \times T \times s}{r \times f}$$

P = working pressure in pounds per square inch.
t = thickness of plate in inches.
T = tensile strength of plate in pounds per square inch.
s = percentage of strength of joint compared to solid plate.
r = radius of shell in inches.
f = factor of safety (usually 5).

From this formula it may be seen that the larger the diameter of the vessel the less pressure the vessel can withstand for the same thickness of the plate and same tensile strength.

Example.—A tubular boiler is 72 inches in diameter and 18 feet long, with double-riveted longitudinal seams of 70% strength, constructed of ⅜" steel plates, with a tensile strength of 60,000 pounds per square inch, the factor of safety being 5. What is the safe working pressure?

Solution.—By substituting the above figures in the formula we obtain $P = \frac{3 \times 60{,}000 \times 0.70}{8 \times 36 \times 5} = 87.5$ pounds safe working pressure.

Calculation of Thickness of Plate.

By transposing the above formula the thickness of plate may be calculated:

$$t = \frac{P \times r \times f}{T \times s}$$

Example.—What thickness of plate is needed for a boiler that should carry a pressure of 105 pounds? The boiler has double-riveted longitudinal seams of 70% strength, the inside diameter of the shell is 60 inches, the tensile strength of the plate 60,000 pounds. The factor of safety is 5.

Solution.—$t = \frac{105 \times 30 \times 5}{60{,}000 \times 0.70} = 0.375 = ⅜"$.

The Hartford Steam Boiler Insurance Co. does not approve of the employment of a plate thicker than one-half inch in the construction of externally fired boilers.

Calculation of Tensile Strength.

The above formula may also be transposed so as to permit calculating the tensile strength, thus:

$$T = \frac{P \times r \times f}{t \times s}$$

Calculation of Diameter of Boiler.

The diameter of the boiler varies with the efficiency of the joint used in the longitudinal seams as will readily be seen from the examples below. The formula for finding the radius of the boiler, also derived by transposition, is the following:

$$r = \frac{T \times s \times t}{P \times f}$$

Example.—A boiler is made of plates 15/32 inch thick, having a tensile strength of 60,000 pounds, and the joint has 70% the strength of solid plate. The factor of safety is 5, and the boiler should carry 100 pounds pressure. What should be the radius of the boiler?

Solution.—The radius $r = \frac{60{,}000 \times 0.70 \times 15}{100 \times 5 \times 32} = 39.375$ or 39⅜ inches, or diameter = 78 inches.

If, however, we should use a triple-riveted butt-joint, having 86% efficiency, then the radius of the boiler should be:

$r = \frac{60{,}000 \times 0.86 \times 15}{100 \times 5 \times 32} = 48.375$ inches, or the diameter = 96 inches.

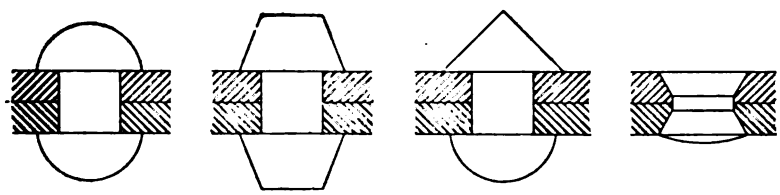


Figure 74. Rivets.

Rivets.

Rivets are used for permanently fastening two or more metal plates together. The plates may be either punched or drilled. With punched, there is generally a loss in strength, especially in plates of more than ½ inch thickness, though this may be partly overcome by subsequent annealing. Drilling causes no appreciable loss in strength.

The rivet may be applied in various forms, and either by hand or by machine, the latter being of course the better method.

Some of the common forms, shown in Figure 74 in the order named, are the snap or button-head, pan-head, steeple-head, and countersunk.

There seems to be a difference of opinion as to whether steel rivets or wrought iron rivets should be used for riveting steel plates. Recent tests, however, seem to show that there is no disadvantage in using steel rivets, in fact the joint is considered even stronger than when using wrought iron rivets.

Strength of Riveted Joints.—There are various styles of riveting, and the strength of joint depends largely on the method used. For ordinary purposes the per cent strengths as compared to the solid plate are taken as follows:

Single-riveted lap-joint—56 per cent.
Double-riveted lap-point—70 per cent.
Double-riveted butt-joint—80 per cent.
Triple-riveted butt-joint—87 per cent.
Quadruple-riveted butt-joint—93 per cent.

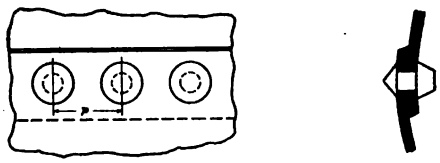


Figure 75. Single-Riveted Lap Joint.

These, of course, vary, as is shown by the following brief description of the different styles with calculations of their strengths. The joints described below are figured on mild steel plates with a tensile strength of 55,000 pounds per square inch, and wrought iron rivets with a shearing strength of 38,000 pounds per square inch.

Single-Riveted Lap-Joint.—This joint, shown in Figure 75, may give way in any of the following ways:

1) Tearing plate along center line of rivets, commonly called tearing net plate.
2) Shearing the rivets.
3) Tearing plate through margin.
4) Shearing plate in front of rivet.
5) Crushing plate or rivet at surface of contact.

In all the calculations and tables below, the following notation is used:

$p =$ pitch of rivets.
$t =$ thickness of plate.
$d =$ diameter of rivet-hole.

The calculation of the strength of this joint follows:

Resistance of net plate to tearing is:

$$(p - d) \times t \times 55{,}000.$$

Shearing strength of the rivet is:

$$0.7854 \times d^2 \times 38{,}000.$$

In a properly designed joint, 3, 4, and 5 need not be considered in figuring the strength. If plates or rivets of other strengths than these are used, the figures may be substituted and the calculations made accordingly. If steel rivets are used, a shearing strength of 44,000 pounds per square inch is generally used.

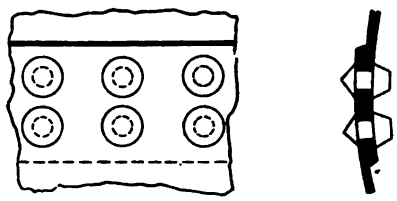


Figure 76. Double-Riveted Lap Joint. (Chain Riveting.)

The strength of the solid plate is:

$$p \times t \times 55{,}000.$$

The per cent strength of net plate compared to the solid plate is:

$$\frac{(p - d) \times t \times 55{,}000 \times 100}{p \times t \times 55{,}000} = \frac{(p - d) \times 100}{p}$$

The per cent strength of the rivet compared to the solid plate is:

$$\frac{0.7854 \times d^2 \times 38{,}000 \times 100}{p \times t \times 55{,}000}$$

Double-Riveted Lap-Joint.—The rivets in this joint may be either chain-riveting (Figure 76) or zig-zag riveting (Figure 77). There is little difference as to strength between these two

methods, though the chain-riveting is considered by some as being slightly stronger than the zig-zag.

This joint may give way in the same manner as a single-riveted lap-joint. The strength is calculated as follows:

1) Strength of net plate is:

$$(p - d) \times t \times 55{,}000.$$

2) Strength of rivets is:

$$2 \times 0.7854 \times d^2 \times 38{,}000.$$

Strength of solid plate is:

$$p \times t \times 55{,}000.$$

Per cent strength of net plate compared to solid plate is:

$$\frac{(p - d) \times t \times 55{,}000 \times 100}{p \times t \times 55{,}000} = \frac{(p - d) \times 100}{p}$$

Per cent strength of rivets compared to solid plate is:

$$\frac{2 \times 0.7854 \times d^2 \times 38{,}000}{p \times t \times 55{,}000}$$

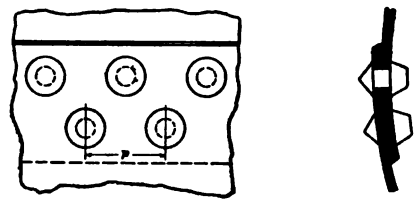


Figure 77. Double-Riveted Lap Joint. (Zig-zag Riveting.)

Example.—In a double-riveted lap-joint, steel plates and wrought iron rivets, the thickness of plate (t) is ½ inch, diameter of rivets (d) is $1^1/_{16}$ inches, and the pitch of rivets (p) is 3½ inches. What is the per cent strength of this joint compared to the solid plate?

Solution.—The per cent strength of the net plate is:

$$\frac{(p - d) \times 100}{p} = \frac{(3\frac{1}{2} - 1^1/_{16}) \times 100}{3\frac{1}{2}} =$$

$$\frac{\left(\frac{56}{16} - \frac{17}{16}\right) \times 100}{\frac{56}{16}} = \frac{39 \times 100}{56} = 69.8\%.$$

The per cent strength of the rivets is:

$$\frac{2 \times 0.7854 \times d^2 \times 38{,}000 \times 100}{p \times t \times 55{,}000} =$$

$$\frac{2 \times 0.7854 \times \frac{17}{16} \times \frac{17}{16} \times 38{,}000 \times 100}{3\frac{1}{2} \times \frac{1}{2} \times 55{,}000} =$$

$$\frac{2 \times 0.7854 \times 17 \times 17 \times 38{,}000 \times 100 \times 2 \times 2}{16 \times 16 \times 7 \times 1 \times 55{,}000} = 70\%.$$

The per cent strength of the joint is, of course, the smaller of these two figures, i. e., 69.8%.

Lap-Joint with Inside Cover-Plate.—This form of riveting is applicable both to single and to double-riveting. They are more efficient than the ordinary lap-joints, but are more expensive and not as satisfactory as the butt-joint.

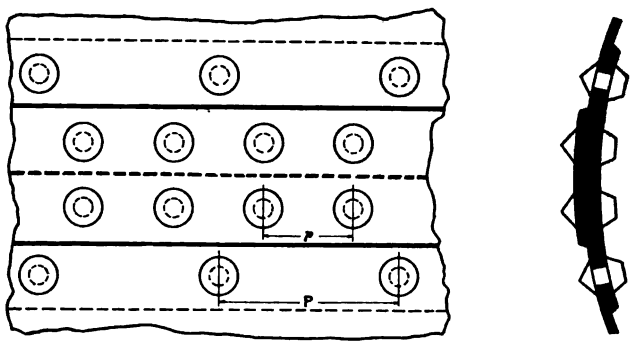


Figure 78. Double-Riveted Butt-Joint.

Double-Riveted Butt-Joint.—This joint, illustrated by Figure 78, in which the two edges of the boiler plate butt together, is held by two straps, an inner and an outer one. There are two rows of rivets on each side, the inner row going through both straps and the outer row going through the inner strap only. The outer row of rivets has double the pitch of the inner row. The inner rivets are in double shear, while the outer rivets are only in single shear. In the calculations double shear will be considered as twice the strength of single shear, and tests seem to bear this out, though some use a figure of from 1.7 to 1.9 instead of 2.

The following is the method of figuring the strength of this joint, referring to Figure 78 for explanation of letters.

1) Resistance of outer net plate to tearing is:

$$(2p - d) \times t \times 55{,}000.$$

2) Shearing strength of two rivets in double shear and one in single shear is:

$$5 \times 0.7854 \times d^2 \times 38{,}000.$$

3) Shearing strength of outer rivet plus resistance of inner net plate to tearing is:

$$0.7854 \times d^2 \times 38{,}000 + (2p - 2d) \times t \times 55{,}000.$$

Strength of solid plate is:

$$2p \times t \times 55{,}000.$$

The per cent strength or efficiency of this joint is calculated as follows:

Per cent strength of outer net plate is:

$$\frac{(2p - d) \times t \times 55{,}000 \times 100}{2p \times t \times 55{,}000} = \frac{(2p - d) \times 100}{2p}$$

Per cent strength of rivets is:

$$\frac{5 \times 0.7854 \times d^2 \times 38{,}000 \times 100}{2p \times t \times 55{,}000}$$

Per cent strength of outer rivet plus inner net plate is:

$$\frac{0.7854 \times d^2 \times 38{,}000 \times 100}{2p \times t \times 55{,}000} + \frac{(2p - 2d) \times t \times 55{,}000 \times 100}{2p \times t \times 55{,}000}$$

As a rule the per cent strength of the outer net plate may be considered as the efficiency of the joint, the per cent strength of the rivets being much higher.

Triple-Riveted Butt-Joint.—In this joint, as with the double-riveted, there are two cover-straps. There are three rows of rivets on each side; two of them pass through the plate and both straps, and one row through the plate and the inner strap only (see Figure 79). The outer row of rivets may also pass through the plate and both straps.

The strength of this joint is figured as follows:

Resistance of outer net plate to tearing is:

$$(2p - d) \times t \times 55{,}000.$$

Shearing strength of all rivets, four in double shear and one in single shear is:

$$9 \times 0.7854 \times d^2 \times 38{,}000.$$

Shearing strength of outer rivet plus resistance of inner net plate is:

$$0.7854 \times d^2 \times 38{,}000 + (2p - 2d) \times t \times 55{,}000.$$

Strength of solid plate $= 2p \times t \times 55{,}000.$

Per cent strength of outer net plate is:

$$\frac{(2p - d) \times t \times 55{,}000 \times 100}{2p \times t \times 55{,}000} = \frac{(2p - d) \times 100}{2p}$$

Per cent strength of rivets is:

$$\frac{9 \times 0.7854 \times d^2 \times 38{,}000 \times 100}{2p \times t \times 55{,}000}$$

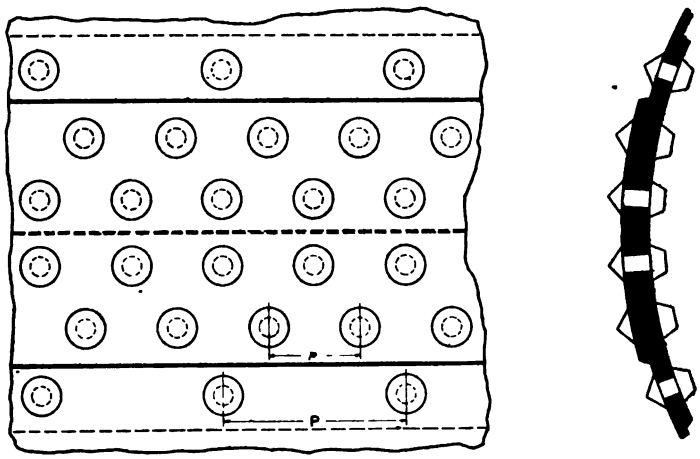


Figure 79. Triple-Riveted Butt-Joint.

Per cent strength of outer rivets plus inner net plate is:

$$\frac{0.7854 \times d^2 \times 38{,}000 \times 100}{2p \times t \times 55{,}000} + \frac{(2p - 2d) \times t \times 55{,}000 \times 100}{2p \times t \times 55{,}000}$$

With the triple-riveted butt-joint, as with the double-riveted, the efficiency or per cent strength may be based on the per cent strength of the outer net plate.

Quadruple-Riveted Butt-Joint.—This form of joint is very strong, having four rows of rivets on each side, two of which pass through the boiler-plate and both straps, and two through plate and inner strap only. The pitch of the rivets in the extreme outer row is four times the pitch of the inner rows,

and that of the next row twice the pitch of the inner rows. It has a strength of 93 to 94 per cent of the solid plate.

The following tables give the dimensions and efficiencies of various riveted joints. They are calculated for steel plates with a tensile strength of 55,000 pounds per square inch, and wrought iron rivets with a shearing strength of 38,000 pounds per square inch.

Single-Riveted Lap-Joints.

Thickness of Plate. Inches.	Diameter of Rivets. Inches.	Pitch of Rivets. Inches.	Efficiency in Per Cent.
1/4	11/16	1 11/16	58.6
5/16	3/4	1 3/4	55.8
3/8	7/8	2	55.4
7/16	15/16	2 1/16	52.8
1/2	1	2 1/16	51.5
9/16	1 3/16	2 3/16	53.1

Double-Riveted Lap-Joints.

Thickness of Plate. Inches.	Diameter of Rivets. Inches.	Pitch of Rivets. Inches.	Efficiency in Per Cent.
1/4	11/16	2 3/4	74.6
3/8	7/8	3 1/16	71.4
7/16	15/16	3 1/8	69.8
1/2	1 1/16	3 1/2	69.6
9/16	1 1/8	3 3/16	68.4
5/8	1 1/8	3 5/16	66.0

Double-Riveted Butt-Joints.

Thickness of Plate. Inches.	Diameter of Rivets. Inches.	Pitch of Rivets. Inches.	Pitch of Outer Rivets. Inches.	Efficiency in Per Cent.
3/8	11/16	2 7/16	4 7/8	83.3
7/16	7/8	2 15/32	4 15/16	82.3
1/2	15/16	2 1/2	5	81.3
9/16	1	2 9/16	5 1/8	80.5
5/8	1 1/16	2 5/8	5 1/4	79.8

Triple-Riveted Butt-Joints.

Thickness of Plate. Inches.	Diameter of Rivets. Inches.	Pitch of Rivets. Inches.	Pitch of Outer Rivets. Inches.	Efficiency in Per Cent.
5/8	7/8	3 3/16	6 3/8	86.2
11/16	1	3 5/8	7 1/4	86.2
1/2	1 1/16	3 7/8	7 3/4	86.2
9/16	1 1/16	4 3/16	8 5/8	86.2
5/8	1 1/4	4 3/16	9 1/8	86.3
11/16	1 3/8	5	10	86.2

Standard Specifications for Special Open-Hearth Plate.

The following standard specifications for special open-hearth plate were adopted by the Association of American Steel Manufacturers, July 17, 1896.

Steel shall be of three grades—Extra Soft, Fire Box, Flange or Boiler.

Extra Soft Steel.—Ultimate strength, 45,000 to 55,000 pounds per square inch. Elastic limit, not less than one-half the ultimate strength. Elongation, 28 per cent. Cold and Quench bends, 180 degrees flat on itself, without fracture on outside of bent portion. Maximum Phosphorus, .04 per cent; maximum Sulphur, .04 per cent.

Fire Box Steel.—Ultimate strength, 52,000 to 62,000 pounds per square inch. Elastic limit, not less than one-half the ultimate strength. Elongation, 26 per cent. Cold and Quench bends, 180 degrees flat on itself, without fracture on outside of bent portion. Maximum Phosphorus, .04 per cent; maximum Sulphur, .04 per cent.

Flange or Boiler Steel.—Ultimate strength, 55,000 to 65,000 pounds per square inch. Elastic limit, not less than one-half the ultimate strength. Elongation, 25 per cent. Cold and Quench bends, 180 degrees flat on itself, without fracture on outside of bent portion. Maximum Phosphorus, .06 per cent; maximum Sulphur, .04 per cent.

Steel for boiler rivets shall be made of extra soft grade specified above.

Test Pieces.—All tests and inspections shall be made at place of manufacture prior to shipment.

The tensile strength, limit of elasticity and ductility shall be determined from a standard test piece cut from the finished material. The standard shape of the test piece for sheared plates shall be as shown in cut. On tests cut from other material the test piece may be either the same as for plates, or it may be planed or turned parallel throughout its entire length. The elongation shall be measured on an original length of eight inches, except in rounds of $\frac{5}{8}$ inch or less in diameter, in which case the elongation shall be measured in a length equal to eight times the diameter of section tested. Four test pieces shall be taken from each melt of finished material; two for tension and two for bending.

Material which is to be used without annealing or further treatment is to be tested in the condition in which it comes from the rolls. When material is to be annealed or otherwise treated before use, the specimen representing such material is to be similarly treated before testing.

Every finished piece of steel shall be stamped with the melt number.

All plates shall be free from surface defects and have a workmanlike finish.

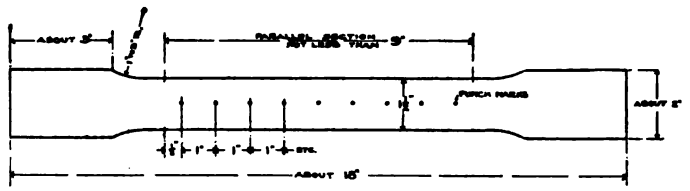


Figure 80. Test-Piece.

Braces and Stays.

The following rules, including those on flat surfaces and flues are largely taken from the General Rules and Regulations of the Board of Supervising Inspectors, Steamboat Inspection Service.

All flat surfaces in a boiler must be stayed. There are numerous methods of staying, depending largely upon the type of boiler and the pressure.

The following formula may be used to calculate the allowable pressure on the stay:

$$P = \frac{6{,}000 \times A}{a}$$

P = safe working pressure in pounds per square inch.
A = least cross-sectional area of stay in square inches.
a = area of surface supported by one stay in square inches.

The stayed portion may be a segment of a circle, as in a horizontal tubular boiler, or it may be rectangular, as in a fire-box boiler. In calculating the area to be stayed, an allowance of 2 inches from the edge is made, as also 3 inches above top row of tubes, as these parts are considered sufficiently supported.

The following formula may be used for calculating the area of a *segment* to be stayed:

$$a = \frac{4 \times h^2}{3} \times \sqrt{\frac{2 \times r}{h} - 0.608}$$

a = area of segment to be stayed in square inches.
h = distance from tubes to shell, minus 5 inches.
r = radius of boiler, minus 3 inches.

In the case of a horizontal tubular boiler, it is this segment which must be braced, and after having found the area, it is an easy matter to calculate the number of stays.

Allowing a tensile strength of the stays of 6,000 pounds per square inch, the calculation may be made as follows:

Example.—In a 72 in. x 16 ft. boiler, the distance from the top row of tubes to the shell is 25 inches. How many 1¼ inch stays are required, having a tensile strength of 55,000 pounds, if the boiler is to carry 100 pounds pressure?

Solution.—$a = \frac{4 \times 20 \times 20}{3} \times \sqrt{\frac{2 \times 33}{20} - 0.608} = 875$ sq. inches that must be supported by stays.

The area that one stay can support is

$$a = \frac{1\frac{1}{4} \times 1\frac{1}{4} \times 0.7854 \times 6{,}000}{100} = 73.5 \text{ square inches.}$$

Hence it would require $\frac{875}{73.5} = 12$ stays.

To Determine the Areas of Diagonal and Gusset Stays.—Multiply the area of a direct stay required to support the surface by the slant or diagonal length of the stay; divide this product by the length of a line drawn at right angles to surface supported

to center of palm of diagonal stay. The quotient will be the required area of the diagonal stay.

$$A = \frac{a \times L}{l}$$

A = sectional area of diagonal stay.
a = sectional area of direct stay.
L = length of diagonal stay.
l = length of line drawn at right angles to boiler head or surface supported to center of palm of diagonal stay.

The sectional area of gusset stays, when constructed of triangular right-angled web plates secured to single or double angle bars along the two sides at right angles, shall be determined by formula for diagonal stays, and shall not be less than 10 per cent greater than would be necessary for a diagonal bolt stay.

The diameter of a screw stay shall be taken at the bottom of the thread, provided it is the least diameter of the stay.

For all stays, the least sectional area shall be taken in calculating the stress allowable.

Screw stay bolts of a greater length than 24 inches will not be allowed in any instance, unless the ends of said bolts are fitted with nuts.

The ends of stays which are upset to include the depth of thread shall be thoroughly annealed after being upset.

When the tensile strength is not known, it may be taken at 55,000 pounds for steel, 45,000 pounds for wrought iron, 18,000 pounds for cast iron and 34,000 pounds for copper.

The sectional area of pins to resist double shear and bending, accurately fitted and secured in crow feet, sling, and similar stays, shall be at least equal to eight-tenths of the required sectional area of the brace.

Breadth across each side and depth to crown of eye shall be not less than 0.35 to 0.55 of diameter of pin. In order to compensate for inaccurate distribution the forks should be proportioned to support two-thirds of the load, thickness of forks to be not less than 0.66 to 0.75 of the diameter of pins.

The combined sectional area of rivets used in securing tee irons and crow feet to shell, said rivets being in tension, shall be not less than the required sectional area of brace. To insure a well-proportioned rivet point, rivets shall be of sufficient length

to completely fill the rivet holes and form a head equal in strength to the body of the rivet. All rivet holes shall be drilled. Distance from center of rivet hole to edge of tee irons, crow feet, and similar fastenings shall be so proportioned that the net sectional areas through sides at rivet holes shall equal the required rivet section. Rivet holes shall be slightly countersunk in order to form a fillet at point and head.

Flat Surfaces.—The maximum stress allowable on flat plates supported by stays shall be determined by the following formula:

$$\text{Working pressure} = \frac{C \times T^2}{P^2}$$

T = thickness of plates in sixteenths of an inch.

P = greatest pitch of stays in inches.

C = 112 for screw stays with riveted heads, plates seven-sixteenths of an inch thick and under.

C = 120 for screw stays with riveted heads, plates above seven-sixteenths of an inch thick.

C = 120 for screw stays with nuts, plates seven-sixteenths of an inch thick and under.

C = 125 for screw stays with nuts, plates above seven-sixteenths of an inch thick and under nine-sixteenths of an inch.

C = 135 for screw stays with nuts, plates nine-sixteenths of an inch thick and above.

C = 175 for stays with double nuts having one nut on the inside and one nut on the outside of plate, without washers or double plates.

C = 160 for stays fitted with washers or doubling strips which have a thickness of at least 0.5 of the thickness of the plate and a diameter of at least 0.5 of the greatest pitch of the stay, riveted to the outside of the plates, and stays having one nut inside of the plate, and one nut outside of the washer or doubling strip. For T take 72 per cent of the combined thickness of the plate and washer or plate and doubling strip.

C = 200 for stays fitted with doubling strips which have a thickness equal to at least 0.5 of thickness of the plate reenforced, and covering the full area braced (up to

the curvature of the flange, if any), riveted to either the inside or outside of the plate, and stays having one nut outside and one inside of the plates. Washers or doubling plates to be substantially riveted. For T take 72 per cent of the combined thickness of the two plates.

C = 200 for stays with plates stiffened, with tees or angle bars having a thickness of at least two-thirds the thickness of plate and depth of webs at least one-fourth of the greatest pitch of the stays, and substantially riveted, on the inside of the plates, and stays having one nut inside bearing on washers fitted to the edges of the web, that are at right angles to the plate. For T take 72 per cent of the combined thickness of web and plate.

All stayed surfaces formed to a curve the radius of which is over 21 inches, excepting surfaces otherwise provided for, may be deemed flat surfaces.

When a flat head has a manhole opening, the flange of which is formed from the solid sheet and turned inward to a depth of not less than twice the thickness of the head, the area of the manhole opening may be deducted from the total area of head to be stayed.

Convex Heads.—All plates used as heads, when new and made to practically true circles, and as described below, may be allowed a steam pressure in accordance with the following formula:

$$P = \frac{T \times S}{R}$$

P = steam pressure allowable in pounds per square inch.
T = thickness of plate in inches.
S = one-fifth of the tensile strength.
R = one-half of the radius to which the head is bumped.

Concave Heads.—For concave heads the pressure allowable will be 0.6 of the pressure allowable for convex heads.

To find the radius of a sphere of which the bumped head forms a part, square the radius of head, divide this by the height of bump required; to the result add height of bump, which will equal diameter of sphere, one-half of which will be the required radius.

Flues.

Working pressures and corresponding minimum thicknesses of wall for long, plain, lap-welded steel flues, 7 to 13 inches diameter, subjected to external pressure only, may be determined by the following formula:

$$P = \frac{(T \times 86{,}670) - (1{,}386 \times D)}{D \times F}$$

D = outside diameter of flue in inches.

T = thickness of wall in inches.

P = working pressure in pounds per square inch.

F = factor of safety.

The working pressure allowable on riveted or lap-welded flues over 13 inches in diameter up to and including 28 inches in diameter, made in sections, and subjected to external pressure only, the lengths not to exceed 3½ times the diameter of the flue, may be determined by the following formula:

$$P = \frac{51.5}{D}[(18.75 \times T) - (L \times 1.03)].$$

P = pressure in pounds per square inch.

D = outside diameter of flue in inches.

L = length of flues in inches, not to exceed 3½ diameters of flue.

T = thickness of wall in sixteenths of an inch.

Morison Corrugated Type.—The following formula gives the steam pressure allowed:

$$P = \frac{C \times T}{D}$$

P = pressure in pounds.

T = thickness in inches, not less than five-sixteenths of an inch.

D = mean diameter in inches.

C = 15,600, a constant, determined from an actual destructive test under the supervision of the Board of Supervising Inspectors, when corrugations are not more than 8 inches from center to center, and the radius of the outer corrugations is not more than one-half of the suspension curve.

In calculating the mean diameter of the Morison furnace, the least inside diameter plus 2 inches may be taken as the mean diameter, thus:

Mean diameter = least inside diameter plus 2 inches.

Adamson Type.—When plain horizontal flues are made in sections not less than 18 inches in length, and not less than five-sixteenths of an inch thick, and flanged to a depth of not less than three times the diameter of rivet hole plus the radius at furnace wall (inside diameter of furnace), the thickness of the flanges to be as near the thickness of the body of the plate as practicable.

The radii of the flanges on the fire side must be not less than three times the thickness of plate.

The distance from the edge of the rivet hole to the edge of the flange must be not less than the diameter of the rivet hole, and the diameter of the rivets before driven must be at least one-fourth inch larger than the thickness of the plate.

The depth of the ring between the flanges must be not less than three times the diameter of the rivet holes, and the ring shall be substantially riveted to the flanges. The fire edge of the ring shall terminate at or about the point of tangency to the curve of the flange, and the thickness of the ring must be not less than one-half inch.

The pressure allowed must be determined by the following formula (Adamson Furnaces in section of not less than 18 inches in length):

$$P = \frac{57.6}{D}[(18.75 \times T) - (1.03 \times L)].$$

P = working pressure in pounds per square inch.

D = outside diameter of furnace in inches.

L = length of furnace in inches.

T = thickness of plate in sixteenths of an inch.

Cylindrical riveted flues and furnaces made in sections of not less than 18 inches in length fitted one into the other and substantially riveted; **combustion chambers** for vertical submerged tubular boilers in the shape of a frustum of a cone; the **inside shells of cylindrical jacketed kettles and tanks,** con-

structed to a practically true circle, may be allowed a steam pressure according to the following formula:

$$P = \frac{51.5}{D}[(18.75 \times T) - (1.03 \times L)].$$

P = working pressure in pounds per square inch.
D = outside diameter of furnace in inches, or outside mean diameter of cone top in inches.
L = length of furnace or flue in inches.
T = thickness of furnace or cone top in sixteenths of an inch, not to be less than five-sixteenths of an inch.

When the diameter of plain furnace, and flues used in vertical type of boilers, the mean diameter of cone tops or the inside shells of cylindrical jacketed kettles and tanks exceeds 42 inches, they may be deemed a flat surface and must be stayed in accordance with rules governing flat surfaces. If a greater working pressure than given by formula is desired for mean diameters under 42 inches, the flues, cone tops or jacketed shells must be substantially stayed for such additional pressure.

Plain Flues (Supported By Angle Bars).—The following formula gives the steam pressure allowed:

$$P = \frac{C \times T}{L \times D}$$

P = pressure in pounds per square inch.
T = thickness in inches, not less than five-sixteenths.
L = length of flue or length between angle rings in feet.
D = outside diameter of flue in inches.
C = 89,600, a constant, when rings are used constructed of angle bars having webs at least 2½ inches wide and a thickness of at least one and one-half times the thickness of the material in the flue; rings to be properly fitted to the flues by thimbles which are not less than 1 inch long and substantially riveted to the flue through the thimble, with rivets pitched not to exceed a distance on the circle in accordance with the rule for flat surface, using the square of the pitch on the circle to determine the stress. The size of the rivets to be determined by the square of the pitch on the circle, using 6,000 pounds per square inch of section for the rivet.

Limiting Formula.—Plain flues, Adamson flues, and flues supported by angle bars, when used as furnaces, should not be allowed a greater working pressure than found by the limiting formula, i. e.:

$$P = \frac{C \times T}{D}$$

P = pressure in pounds per square inch.
C = 8,000, a constant.
T = thickness of material in inches.
D = outside diameter of flue in inches.

Energy Stored Up in Steam Boilers.

The following figures give a general idea of the amount of energy stored up in a boiler. The boiler may be assumed to be of 60 H. P. and to contain the given quantities of water and steam at 75 pounds gauge pressure.

Weight of empty boiler	9,500 pounds
Weight of water in boiler	8,300 pounds
Weight of steam in boiler	21 pounds
Total weight	17,821 pounds
Energy available in water	50,008,790 foot-pounds
Energy available in steam	1,022,730 foot-pounds
Total available energy	51,031,520 foot-pounds
Energy per pound of total weight including boiler	2,865 foot-pounds

The height to which the empty boiler could be thrown by this stored up energy may be found, assuming an initial velocity of 588 feet per second, by dividing the total number of foot-pounds by the weight of the boiler:

$$\frac{51{,}031{,}520}{9{,}500} = 5{,}371$$

The boiler could be thrown to a height of 5,371 feet, or over a mile.

Boiler Explosions.

An explosion of a boiler may be caused by any of the following: *low water,* causing overheating and consequent weakening of the boiler sheet or wall; *defective construction; exces-*

sive pressure; crystalline structure of the steel; *defective braces,* etc.; *sudden release of pressure* due to opening a large valve too suddenly.

A boiler should be inspected and tested at least once a year. A boiler should be tested hydrostatically with a pressure not exceeding 50% more than the safe working pressure. Once every year a boiler should receive what is called a hammer test, which means a thorough examination of the boiler externally and internally in connection with all the auxiliary appliances. The sheets are examined to determine any possible corrosion, blistering, bagging, pitting, loose or defective rivets and braces, uncleanliness, etc. The feed-water pipe, blow-off connections, safety valve, water column, steam gauge, etc., are also examined as to whether they are able to perform their proper functions. This is usually done by the steam boiler insurance companies.

PERFORMANCE OF BOILERS.

The performance of a steam boiler embraces its capacity for generating steam and its economical use of fuel. The capacity is dependent on the grate surface, heating surface, kind of coal burned and strength of draft. Economy in the use of fuel depends upon the kind of combustion that is obtained with the coal in the furnace, upon the proper regulation of air supply, i. e., upon the ratio of air supplied to the amount of air consumed, and upon the capacity of the boiler for absorbing the heat from the furnace.

This absorption of heat depends upon the relation of the heating surface to the amount of coal burned in a given time; upon the arrangement of the gas passages and upon the cleanliness of the surface. The economy may increase with the capacity, or the capacity may increase at the expense of economy. The latter is generally due to overcrowding of the boiler, causing the gases to pass into the chimney at too high a temperature.

Heating Surface of Boiler.

The heating surface of a boiler is that part of its surface which comes in contact with the hot gases. In tubular boilers, two-thirds of the cylindrical shell, the entire surface of the tubes, and two-thirds of both heads, minus the tube area, are exposed to the hot gases. If we take, for example, a boiler 60 inches in

diameter and 18 feet long, with 46 four-inch tubes, its heating surface in square feet would be as follows:

Surface of shell:

$$\frac{60 \times 3.1416 \times 18 \times 2}{12 \times 3} = 188.496 \text{ sq. ft.}$$

Internal tube surface:

$$\frac{3.732 \times 3.1416 \times 46 \times 18}{12} = 808.985 \text{ sq. ft.}$$

(3.732 is the internal diameter of a 4-inch tube).*

Surface of each head:

$$\frac{60 \times 60 \times 0.7854 \times 2}{144 \times 3} = 13.09 \text{ sq. ft.}$$

Surface of flue heads:

$$\frac{3.732 \times 3.732 \times 0.7854 \times 46}{144} = 3.4937 \text{ sq. ft.}$$

$13.09 - 3.4937 = 9.5963$ sq. ft.

$9.5963 \times 2 = 19.1926$ sq. ft. for both heads, hence the total heating surface is:

188.496	shell
808.985	tubes
19.193	heads
1,016.674	sq. ft.

Relation of Grate Surface, Heating Surface, Fuel and Water.

It is well known that in a given boiler with a given furnace, the greater the quantity of fuel consumed per hour, the greater also the amount of water evaporated per hour. The quantity of water evaporated, however, increases at a less rate than the fuel consumed, that is to say, the quantity of water evaporated per pound of fuel is diminished. This decrease in efficiency is obviously due to a larger quantity of heat being wasted by escaping through the chimney, as indicated by the higher temperature of the gases, which remain unused for evaporation. For maximum economy in using any kind of coal, a boiler should be proportioned so that at least one square foot of heating surface is allowed for every 3 pounds of water evaporated from and at 212°F. per hour.

* For a water-tube boiler, the external diameter should be used.

For good economy with a medium grade of coal, the ratio of grate area to heating surface may be taken 1 to 40 for tubular boilers, and 1 to 45 for water-tube boilers or even 1 to 50 with sharp draft. With chain grates the ratio is sometimes taken as 1 to 60 for water-tube boilers.

If the heating surface of a tubular boiler is divided by 12, it expresses its rated horse-power. The rated horse-power of water-tube boilers is found by dividing the surface coming in contact with the hot gases by 10, instead of 12. Thus, for example, the rated horse-power of a tubular boiler of the size mentioned above, which was found to have 1,016 square feet of heating surface, would be $\frac{1,016}{12} = 85$ horse-power. In the case of a water-tube boiler with the same amount of heating surface, the boiler would be rated at $\frac{1,016}{10} =$ approximately 100 horse-power.

Dimensions of Boiler Feed Pumps, Injectors and Feed-Water Heaters.

The size of the boiler feed pump or injector depends on the heating surface of the boiler or battery of boilers. The pump or injector should be able to deliver into the boiler 7 pounds of water per square foot of heating surface per hour, and should be able to do this work at a normal speed. Again applying the above calculated boiler to this rule, we require a pump of a capacity of $1,016 \times 7 = 7,112$ pounds of water per hour, and assuming that one gallon of water weighs 8 pounds, the capacity of the pump per minute would have to be $\frac{7,112}{60 \times 8} =$ approximately 15 gallons.

The size of the heater, if of the closed type, is also dependent on the amount of heating surface. One square foot of heater surface should be allowed for every 20 square feet of boiler heating surface, which would require a closed heater for the above calculated boiler, which contains $\frac{1,016}{20} = 51$ square feet of water heating surface.

Actual Boiler Horse-Power.

A boiler horse-power is an established unit with which the capacity of a boiler for evaporating water is measured, and it

would be a rather inaccurate quantity had not a certain standard been generally adopted. The recognized commercial boiler horse-power is the capacity to evaporate 34.5 pounds of water per hour from a feed-water temperature of 212°F., into steam of 212°F. Based on the new steam tables of Marks & Davis, this is equal to evaporating 30.018 pounds of water from a feed-water temperature of 100°F., into steam of 70 pounds gauge pressure, and corresponds to 33,479 B. T. U. absorbed by the water in the steam boiler in one hour.

Unit of Evaporation.

The quantity of heat necessary to evaporate one pound of water of 212°F. into steam of 212°F. has been adopted as the unit of evaporation. This is equal to the latent heat of evaporation and amounts to 970.4 B.T.U.

Equivalent Evaporation.

Since various feed-water temperatures and various pressures are used in a boiler, and since the horse-power of a boiler is expressed in terms of water evaporated from a feed-water temperature of 212°F. to steam of 212°F., it becomes desirable to express the evaporation in these terms. This is commonly called the equivalent evaporation from and at 212°F.

It is expressed by the following formula:

$$E = W \times \frac{H - (t - 32)}{L}$$

E = equivalent evaporation from and at 212°F.

W = pounds of water actually evaporated.

H = total heat in steam above 32°F. in B.T.U.

t = temperature of feed-water in °F. before entering boiler.

L = 970.4 = latent heat of steam at 212°F. in B.T.U. per pound.

The following example shows the application to the calculation of *boiler horse-power.*

Example.—A boiler evaporates 47,995 pounds of water from 182°F. into dry steam of 100 pounds gauge pressure in 12 hours. How many actual horse-power does it develop?

Solution.—H. P. $= \frac{47{,}995 \times [1{,}188.8 - (182 - 32)]}{34.5 \times 12 \times 970.4}$

$= 124$ H. P.

Factors of Evaporation.

Feed water temp.	Gauge Pressure										
	0	20	40	60	80	100	120	140	160	180	200
212	0.0000	1.0168	1.0257	1.0315	1.0360	1.0396	1.0425	1.0449	1.0469	1.0487	1.0503
210	1.0021	1.0189	1.0278	1.0336	1.0381	1.0417	1.0446	1.0470	1.0490	1.0508	1.0524
200	1.0124	1.0292	1.0381	1.0440	1.0484	1.0520	1.0549	1.0574	1.0593	1.0612	1.0627
190	1.0227	1.0395	1.0484	1.0543	1.0587	1.0623	1.0652	1.0677	1.0696	1.0715	1.0730
180	1.0331	1.0499	1.0588	1.0647	1.0691	1.0727	1.0756	1.0780	1.0800	1.0818	1.0834
170	1.0434	1.0602	1.0691	1.0750	1.0794	1.0830	1.0859	1.0883	1.0903	1.0921	1.0937
160	1.0537	1.0705	1.0794	1.0853	1.0897	1.0933	1.0962	1.0986	1.1006	1.1024	1.1040
150	1.0640	1.0808	1.0897	1.0956	1.1000	1.1036	1.1065	1.1089	1.1109	1.1128	1.1143
140	1.0743	1.0911	1.1000	1.1059	1.1103	1.1139	1.1168	1.1192	1.1212	1.1231	1.1246
130	1.0846	1.1014	1.1102	1.1161	1.1206	1.1242	1.1270	1.1295	1.1315	1.1333	1.1349
120	1.0949	1.1117	1.1205	1.1264	1.1309	1.1345	1.1373	1.1398	1.1418	1.1436	1.1452
110	1.1052	1.1220	1.1308	1.1367	1.1411	1.1447	1.1476	1.1501	1.1521	1.1539	1.1555
100	1.1154	1.1322	1.1411	1.1470	1.1514	1.1550	1.1579	1.1603	1.1623	1.1642	1.1657
90	1.1257	1.1425	1.1514	1.1573	1.1617	1.1653	1.1682	1.1706	1.1726	1.1745	1.1760
80	1.1360	1.1528	1.1617	1.1675	1.1720	1.1756	1.1784	1.1809	1.1829	1.1847	1.1863
70	1.1463	1.1631	1.1719	1.1778	1.1822	1.1858	1.1887	1.1912	1.1931	1.1950	1.1966
60	1.1566	1.1734	1.1822	1.1881	1.1925	1.1961	1.1990	1.2015	1.2034	1.2053	1.2069
50	1.1669	1.1837	1.1925	1.1984	1.2028	1.2064	1.2093	1.2118	1.2137	1.2156	1.2171

This is worked as follows: The equivalent evaporation, or the pounds of water which this boiler evaporates from and at 212°F. is found from the formula given above, as follows:

$$E = \frac{47{,}995 \times [1{,}188.8 - (182 - 32)]}{970.4}$$

To find the evaporation per hour, this is divided by 12, the number of hours required for the above evaporation. The horsepower is then of course this figure divided by 34.5 (see page 234).

Factor of Evaporation.

It is easier, however, to establish a so-called *"factor of evaporation"* for each pound of water evaporated. This is done according to the formula given above, i. e., $\frac{H - (t - 32)}{L}$

This gives a so-called multiplier, or a factor of evaporation, by which the water actually evaporated in one hour is multiplied, the result being then divided by 34.5, which expresses the horsepower the boiler develops.

Example.—Using the figures of the preceding example gives as the factor of evaporation: $\frac{1{,}188.8 - (182 - 32)}{970.4}$ or 1.07049.

By multiplying the pounds of water evaporated in one hour, i. e., $\frac{47{,}995}{12} = 3{,}999.6$ pounds, by this factor, and dividing the result by 34.5, we obtain: $\frac{3{,}999.6 \times 1.07049}{34.5} = 124$ H. P.

In order to avoid the trouble of calculating the factor of evaporation in each case, tables have been compiled, which contain the factors for different feed-water temperatures and gauge pressures. The table on page 235 is based on the new steam tables of Marks and Davis.

Efficiency of Boiler.

In speaking of the efficiency of the boiler, reference may be had either to the efficiency of the boiler and grate, or to the efficiency of the boiler alone. By *efficiency of boiler and grate* is meant the ratio between the heat units in the steam produced per pound of dry coal fired and the heat units contained in one pound of dry coal. By *efficiency of boiler* alone is meant the

ratio between the heat units in the steam produced per pound of combustible burned on the grate and the heat units in one pound of combustible.

The *efficiency of the grate* alone is the ratio between the efficiency of the boiler and grate and the efficiency of the boiler.

Expressed in formulas, we have:

$$\text{Efficiency of boiler and grate} = \frac{\text{heat absorbed per lb. dry coal } \textit{fired}}{\text{calorific value of 1 lb. dry coal}}$$

$$\text{Efficiency of boiler} = \frac{\text{heat absorbed per lb. combustible } \textit{burned}}{\text{calorific value of 1 lb. combustible}}$$

$$\text{Efficiency of grate} = \frac{\text{Efficiency of boiler and grate}}{\text{Efficiency of boiler}}$$

The following example gives the general method of calculation. For further details, see under "Boiler Tests."

Example.—A boiler evaporates 8 pounds of water from and at 212° F. per pound of coal fired. The coal contains 12% of moisture, 7% ash and 81% combustible. The total refuse in the ash pit is 11.5% of the coal fired. The theoretical heating value per pound of coal as fired is 11,400 B.T.U. What is the efficiency of boiler and grate; efficiency of boiler; efficiency of grate?

Solution.—The heating value of one pound of dry coal is:

$$\frac{11{,}400}{1.00 - 0.12} = \frac{11{,}400}{0.88} = 12{,}955 \text{ B. T. U.}$$

Heating value per pound of combustible is:

$$\frac{11{,}400}{1.00 - (0.12 + 0.07)} = \frac{11{,}400}{0.81} = 14{,}074 \text{ B. T. U.}$$

Amount of water evaporated from and at 212°F. per pound of dry coal is:

$$\frac{8}{0.88} = 9.09 \text{ pounds.}$$

Heat absorbed per pound of dry coal is:

$$9.09 \times 970.4 = 8{,}821 \text{ B.T.U.}$$

Combustible burned on the grate is:

$$100 - (12 + 11.5) = 100 - 23.5 = 76.5\%.$$

Amount of water evaporated from and at 212°F. per pound of combustible burned on the grate is:

$$\frac{8}{0.765} = 10.46 \text{ pounds.}$$

Heat absorbed per pound of combustible burned on the grate is:

$$10.46 \times 970.4 = 10{,}150 \text{ B.T.U.}$$

From the formulas given and data calculated, the efficiencies may now readily be calculated:

$$\text{Efficiency of boiler and grate} = \frac{8{,}821 \times 100}{12{,}955} = 68.09\%.$$

$$\text{Efficiency of boiler alone} = \frac{10{,}150 \times 100}{14{,}074} = 72.12\%.$$

$$\text{Efficiency of grate} = \frac{68.09 \times 100}{72.12} = 94.41\%.$$

The efficiency of the boiler and grate being about 68%, there is a loss of about 32%. A loss of 40 to 50% is not unusual, but on the contrary very frequent, while a loss of less than 20% is seldom if ever attained.

Combustion is always more or less incomplete, as fuel is lost by dropping through the grate, and considerable heat is carried away by the hot gases and by radiation. When the difference between the theoretical and actual values seems too small to cover these losses, there is good cause for hesitancy in accepting the accuracy of the figures, which would represent an abnormal rate of evaporation.

Boiler Tests.

The object of a boiler test is to determine the amount of water that can be evaporated under existing conditions by a certain kind of coal in a given time. A boiler test may be made either for determining the capacity or the economy of the boiler.

Methods of Testing Boilers.*

There are two methods of starting and stopping a test, the so-called *standard* method and the *alternate* method. The latter is employed where it is inconvenient to make use of the standard method.

* The following methods are taken largely from the reports of the Committee on Boiler Trials, American Society of Mechanical Engineers.

The *standard* method of starting and stopping a test is to raise steam to the working pressure, remove all the fire from the grate rapidly, close the damper, clean the ash-pit, and then as quickly as possible, start a new fire with weighed wood and coal, noting the time and the water level, just before lighting the fire. At the end of the test, the whole fire, which has been burned low, is removed, the grates and ash-pit cleaned, and a record taken of the time of hauling the fire. The water level should as far as possible be the same at the close as at the beginning of the test.

With the *alternate* method of starting and stopping a test, the boiler must have been thoroughly heated by a preliminary run. The fires are then burned low, and well cleaned. The amount of coal left on the grate is recorded as nearly as it can be estimated, and record made of the pressure of steam, of the water level and the starting time. Fresh coal previously weighed is now fired, and the ash-pit thoroughly cleaned at once after starting. Before the end of the test the fire is burned low, to the same point as before the start, and cleaned in such a manner as to leave a bed of coal on the grates, of the same depth and in the same condition as at the start. When this stage is reached, the time is noted, and recorded as the stopping time. The water level and steam pressure should previously have been brought as nearly as possible to the same point as when beginning the test. If the water level is not the same as at the start, a correction should be made by computation, but not by operating the pump after the test is completed. Uniformity of conditions as to the pressure of steam, the height of water, the rate of evaporation, the thickness of the fire, the time of firing, the quantity of coal fired at one time and the intervals between the times of cleaning the fire, should prevail. The method of firing to be carried on in such tests should be directed by the expert or the person in charge of the test, and the method adopted should be strictly adhered to by the fireman throughout the test. The dimensions, position, etc., of grate, heating surface, flues, and chimney; the proportionate amount of air space openings in grate; the kind of draft (natural or forced), should all be measured, or noted and recorded. In all important tests, a sample of

coal should be selected for chemical analysis or calorimeter examination.

Taking of Coal Sample.—In order that the results from the analysis of the coal be reliable, it is important that the coal sample be taken properly. During the test, a sample of the coal should be taken from each weighed portion furnished to the fireman. After the test is over, these samples should be thoroughly mixed and then spread out evenly on the cleaned floor in the form of a square. This is divided into four portions, three of which are thrown back to the bulk of the coal, while the fourth is pounded into small pieces and again thoroughly mixed and evenly spread out. This is once more divided into four parts, one of which is again broken into smaller pieces, mixed, spread out and divided as before. This is repeated until the last portion amounts to about two pounds. This last quantity is divided into two parts, each of which is put into a receptacle which can be closed tightly, a mason jar for instance, and one is sent to the chemist for analysis, while the other one is reserved for future reference.

Water meters should be used as a rule only as a check upon other measurements, or in case comparative tests are made on the same boiler on different days with different kinds of fuel, for determination of the value of the fuel. For accurate work the water should be weighed or measured in a tank.

Thermometers should be inserted for taking the temperatures of the feed-water, flue gases and boiler room, and a **draft gauge** should be used for measuring the draft.

The steam gauge of the boiler should be tested as to its correctness. The weight of the coal and ashes should each be carefully taken and recorded.

Before beginning a test, all superfluous pipes and connections, including the blow-off, should be disconnected, or stopped with a blank flange, and any unauthorized opening of valves should be guarded against. If an injector is used, it must receive the steam directly from the boiler under test. The steam pipe should be so arranged that the water of condensation cannot return to the boiler, and that it is trapped if necessary.

The coal should be weighed and delivered to the fireman in equal portions, each sufficient for an hour's run, and a fresh

portion should not be delivered until the previous one has all been fired, the time required to consume each portion being noted. At the same time, the amount of water fed into the boiler, the height of the water, the steam pressure and the average feed-water temperature should be accurately noted and recorded. By thus recording the amount of water evaporated by each successive portion of coal, the record of the test may be subdivided if desired. This permits discovering at the end of the test, the degree of uniformity of combustion at the different stages of the test.

Calorimeter tests should be made to ascertain the percentage of moisture in the steam or the degree of superheating. At least ten such tests should be made, and the greatest care should be taken in the measurement of weights and temperature.

In tests made for commercial purposes, certain observations and measurements which would be necessary in case of scientific research, need not be made. These are the measurement of the air supply, the determination of its moisture content, the measurement and analysis of the flue gases, the determination of the amount of heat lost by radiation, of the amount of infiltration of air through the setting, the direct determination by calorimeter tests of the absolute heating value of the fuel, and (by condensation of all steam made in the boiler) the determination of the total heat imparted to the water.

The analysis of the flue gases is an especially valuable method for determining the relative value of different methods of firing or of different kinds of furnaces.

Entering Results of Tests.—The final results should be entered upon a properly prepared blank, of which the following is a specimen, and should include all the items adapted for the specific object for which the test is made.

Report Sheet for Boiler Tests.

General Notes and Total Quantities.

1. Date of trial.
2. Duration of trial in hours.
3. Type of boiler.
4. Rated capacity of boiler in horse-power.

5. Kind of furnace.
6. Grate surface in square feet.
7. Heating surface in square feet.
8. Kind of coal.
9. Cost of coal per ton.
10. Total weight of coal consumed in pounds.
11. Percentage of moisture and of ash in coal.
12. Total weight of dry coal consumed.
13. Total ash and refuse.
14. Per cent of ash and refuse.
15. Total amount of water evaporated.
16. Factor of evaporation.
17. Equivalent water evaporated into dry steam from and at 212°F.

Hourly Quantities.

18. Dry coal consumed per hour.
19. Dry coal per square foot of grate surface, per hour.
20. Water evaporated into dry steam, per hour.
21. Equivalent evaporation into dry steam per hour from and at 212°F.
22. Evaporation per hour from and at 212°F. per square foot of heating surface.

Average Pressures, Temperatures, Etc.

23. Steam pressure.
24. Feed-water temperature.
25. Flue gas temperature.
26. Draft between boiler and damper.
27. Percentage of moisture in steam.

Horse-Power.

28. Horse-power developed (34.5 lbs. of water evaporated per hour into dry steam from and at 212°F. equals 1 horse-power).
29. Percentage of builder's rated horse-power developed.
30. Horse-power per square foot of grate surface.
31. Horse-power per square foot of heating surface.

Economic Results.

32. Water apparently evaporated per pound of coal.
33. Equivalent evaporation from and at 212°F. per pound of actual coal.

34. Equivalent evaporation from and at 212°F. per pound of dry coal.

35. Equivalent evaporation per pound of combustible.

Efficiency.

36. Calorific value per pound of actual coal in B.T.U.

37. Calorific value per pound of dry coal in B.T.U.

38. Calorific value per pound of combustible in B.T.U.

39. Efficiency of boiler and grate.

40. Efficiency of boiler alone.

41. Efficiency of grate alone.

Cost of Evaporation.

42. Cost of coal for evaporation of 1,000 pounds of water from and at 212°F.

43. Pounds of dry steam produced from and at 212° F. for $1.00.

In numbers 35 and 38, the term combustible has been used. By this is understood coal free from ash and moisture. For instance if a coal contains 15% of ash and 8% of moisture the percentage of combustible is 100—(15+8) or 77%.

Calculation of Boiler Test.—Only part of the data for the above report sheet are taken from direct observation during the test, a large number of the entries being calculated. It will be shown below which are observed and which are calculated and in what manner.

1. Observed.
2. Observed.
3. Observed.
4. Calculated by dividing the total heating surface of the boiler by 12 for tubular, by 10 for water-tube, and by 8 for Scotch marine boilers.
5. Observed.
6. Calculated by multiplying the width by the length of the grate.
7. Calculated as described on page 231.
8. Observed.
9. Observed.
10. Observed.
11. Found by analysis.

12. Found as follows: $\text{No. } 10 - \frac{\text{No. } 10 \times \text{No. } 11^*}{100}$
13. Observed.
14. Found as follows: $\frac{\text{No. } 13 \times 100}{\text{No. } 10}$
15. Observed.
16. Found as described on page 236 or from table, page 235.
17. Found as follows: $\text{No. } 16 \times \left(\text{No. } 15 - \frac{\text{No. } 15 \times \text{No. } 27}{100}\right)$
18. Found as follows: $\frac{\text{No. } 12}{\text{No. } 2}$
19. Found as follows: $\frac{\text{No. } 18}{\text{No. } 6}$
20. Found as follows: $\left(\text{No. } 15 - \frac{\text{No. } 15 \times \text{No. } 27}{100}\right) \div \text{No. } 2.$
21. Found as follows: $\frac{\text{No. } 17}{\text{No. } 2}$
22. Found as follows: $\frac{\text{No. } 21}{\text{No. } 7}$
23. Observed.
24. Observed.
25. Observed.
26. Observed.
27. Found as described on page 257.
28. Found as follows: $\frac{\text{No. } 21}{34.5}$
29. Found as follows: $100 - \frac{\text{No. } 28 - \text{No. } 4}{\text{No. } 4} \times 100.$
30. Found as follows: $\frac{\text{No. } 28}{\text{No. } 6}$
31. Found as follows: $\frac{\text{No. } 28}{\text{No. } 7}$
32. Found as follows: $\frac{\text{No. } 15}{\text{No. } 10}$
33. Found as follows: $\text{No. } 32 \times \text{No. } 16.$

* Moisture only.

34. Found as follows: $\frac{\text{No. 17}}{\text{No. 12}}$

35. Found as follows: $\frac{\text{No. 17}}{\text{No. 12} - \text{No. 13}}$

36. Found by analysis.

37. Found as follows: $\frac{\text{No. 36} \times 100}{100 - \text{No. 11*}}$

38. Found as follows: $\frac{\text{No. 36} \times 100}{100 - \text{No. 11†}}$

39. Found as follows: $\frac{\text{No. 34} \times 970.4}{\text{No. 37}} \times 100.$

40. Found as follows: $\frac{\text{No. 35} \times 970.4}{\text{No. 38}} \times 100.$

41. Found as follows: $\frac{\text{No. 39} \times 970.4}{\text{No. 40}} \times 100.$

42. Found as follows: $\frac{\text{No. 9}}{\text{No. 33} \times 2}$

43. Found as follows: $\frac{2{,}000 \times \text{No. 33}}{\text{No. 9}}$

The following is a report sheet properly filled out and all entries and calculations made.

Report Sheet for Boiler Test.

General Notes and Total Quantities.

1. Date of trial.
2. Duration of trial in hours.....10 hours.
3. Type of boiler.....Cahall Water-Tube.
4. Rated capacity of boiler in H. P.....300 H. P.
5. Kind of furnace.....Common Grate.
6. Grate surface in square feet.....66 square feet.
7. Heating surface in square feet.....3,000 square feet.
8. Kind of coal used.....Illinois Mine Run.
9. Cost of coal per ton.....$2.00.
10. Total weight of coal consumed, in pounds....13,200 pounds.
11. Percentage of moisture and of ash in coal.....12% and 5%.

* Moisture only.
† Moisture plus ash

12. Total weight of dry coal consumed, in pounds. 11,616 pounds.
13. Total ash and refuse..... 1,215 pounds.
14. Per cent of ash and refuse..... 9.2%.
15. Total amount of water evaporated..... 87,120 pounds.
16. Factor of evaporation..... 1.0602.
17. Equivalent water evaporated into dry steam from and at 212°F. ..... 90,518 pounds.

Hourly Quantities.

18. Dry coal consumed per hour..... 1,161.6 pounds.
19. Dry coal per sq. foot of grate surface, per hour.. 17.6 pounds.
20. Water evaporated into dry steam per hour.... 8,538 pounds.
21. Equivalent evaporation into dry steam per hour from and at 212°F. ..... 9,052 pounds.
22. Evaporation into dry steam per hour from and at 212°F. per square foot of heating surface..... 3.02 pounds.

Average Pressures, Temperatures, Etc.

23. Steam pressure ..... 100 pounds.
24. Feed-water temperature ..... 192°F.
25. Flue gas temperature ..... 440°F.
26. Draft between boiler and damper..... 0.54 inch.
27. Percentage of moisture in steam..... 2%.

Horse-Power.

28. Horse-power developed ..... 262.3 H. P.
29. Percentage of builder's rating..... 87.43%.
30. Horse-power per square foot of grate surface.... 3.97 H. P.
31. Horse-power per sq. foot of heating surface... 0.0874 H. P.

Economic Results.

32. Water apparently evaporated per pound of coal.. 6.6 pounds.
33. Equivalent evaporation from and at 212°F. per pound of actual coal ..... 6.997 pounds.
34. Equivalent evaporation into dry steam from and at 212°F. per pound of dry coal fired..... 7.793 pounds.
35. Equivalent evaporation into dry steam per pound of combustible burned on the grate..... 8.703 pounds.

Efficiency.

36. Calorific value of actual coal per lb. in B.T.U. . 11,400 B.T.U.
37. Calorific value per pound of dry coal in B.T.U. 12,955 B.T.U.
38. Calorific value per pound of combustible in B.T.U. ..... 13,735 B.T.U.
39. Efficiency of boiler and grate ..... 58.37%.
40. Efficiency of boiler alone ..... 61.49%.
41. Efficiency of grate alone ..... 94.93%.

Cost of Evaporation.

42. Cost of evaporation of 1,000 pounds of water from and at 212° F. ..... 14.29 cents.
43. Pounds of dry steam produced from and at 212°F. for $1.00 ..... 6,997 pounds.

Steam and Steam Appliances

The vapor generated by heating water to the boiling point is called steam. In reality it is invisible, and the fact that it can be seen is only due to its partial condensation when coming into contact with cold air.

Saturated Steam.

Saturated steam is steam which is or has been in contact with water, and whose temperature depends upon the pressure at which it is generated, and vice versa. Thus, for every temperature of saturated steam there is only one pressure possible, and for every pressure only one temperature. At a pressure of one atmosphere the temperature is 212° F., and as the pressure becomes higher, the temperature of the steam, and at the same time that of the water in contact with it, increases.

Saturated steam is called *wet steam* in case it contains water in the form of mist or spray in suspension, and *dry steam* if this is not the case. Dry steam, however, may also be superheated steam. A description of the methods used for determining the percentage of moisture in steam will be found on page 257.

Properties of Saturated Steam.—The table of properties of saturated steam, as given on pages 252 to 254 is the new one of Marks and Davis, using as a basis that 1 B. T. U. is 1/180 of the heat required to raise 1 pound of water from 32 to 212° F., and that this heat is equal to 777.52 foot-pounds. This table gives in most cases, somewhat different values than the old table, and is more accurate. Peabody's table uses slightly different standards, and gives results a trifle different than that of Marks and Davis.

Specific Heat of Saturated Steam.—Strictly speaking, there is no specific heat of saturated steam. When a unit weight of saturated steam is increased in temperature and pressure, the

volume decreasing, so as to just keep it saturated, the specific heat is negative, and decreases as the temperature increases.

Determination of Steam Pressure.—The steam pressure may be expressed in two ways. It may either be the pressure above the atmospheric pressure, or the pressure above absolute vacuum. The former is called *gauge pressure,* and the latter *absolute pressure.*

Ordinary pressure gauges, such as are in use on boilers, are employed in the former case, while the absolute pressure can be calculated by adding the atmospheric pressure to that indicated by the pressure gauge.

In removing air from a vessel, the pressure is reduced correspondingly below that of the atmosphere until an absolute vacuum is obtained, in which case there is no pressure. For the determination of pressures below that of the atmosphere, so-called *vacuum gauges* are employed. These are constructed on the principle of the mercury barometer, and indicate the reduction in pressure in inches of mercury. If there is no reduction, the indication is zero; in case of complete vacuum they indicate 30 inches.

A pressure of 14.7 pounds per square inch will balance a mercury column 30 inches in height (temperature of the mercury 58.4° F.). If, therefore, a vacuum gauge indicates 26 inches of vacuum, the absolute pressure can be calculated as follows:

$$14.7 - \frac{26 \times 14.7}{30} = 1.96 \text{ pounds.}$$

On the other hand, an absolute pressure of 3.5 pounds would show on the vacuum gauge:

$$\frac{(14.7 - 3.5) \times 30}{14.7} = 22.86 \text{ inches of vacuum.}$$

Calculation of Temperature and Pressure of Saturated Steam.—Though the relation of pressure and temperature of saturated steam is given in all tables on saturated steam, it may be of interest to know how this can be calculated. The formulas for the calculations are as follows:

$$t = \frac{2{,}938.16}{6.1993544 - \log.p} - 371.85$$

$$\log. p = \frac{6.1993544 \times t - 632.93}{371.85 + t}$$

t = temperature of the steam in degrees Fahrenheit.
p = absolute pressure in pounds per square inch.

The results obtained are accurate between 120 and 446° F. corresponding to absolute pressures of 1.69 and 405 pounds per square inch.

By using approximate figures, both formulas may be considerably abbreviated, while the results obtained are practically identical with those calculated by means of the original formulas. The simplified formulas read as follows:

$$t = \frac{2{,}938.2}{6.2 - \log.p} - 372$$

$$\log.p = \frac{6.2 \times t - 633}{372 + t}$$

Example.—What is the temperature of saturated steam at 85.3 pounds gauge pressure?

Solution.—Since the pressure p in the above formula is the absolute pressure, we have to add 14.7 pounds to the gauge pressure 85.3 in order to obtain the absolute pressure, which will then be 100 pounds. The logarithm of 100 is 2, hence we have:

$$t = \frac{2938.2}{6.2 - 2} - 372 = 327.6° \text{ F.}$$

Example.—What is the absolute pressure corresponding to a steam temperature of 323° F.?

Solution.—$\text{Log. } p = \frac{6.2 \times 323 - 633}{372 + 323} = \frac{1{,}369.6}{695} = 1.97065.$

In the table of logarithms we find that the number corresponding to 1.97065 is 93.5. The absolute pressure of the steam is, therefore, 93.5 pounds. By substracting 14.7 pounds from this, we obtain 78.8 pounds as the gauge pressure.

Latent Heat of Steam.—A certain quantity of heat is required to change water into steam of the same temperature. This heat cannot be measured by means of the thermometer; it is hidden or latent, and is therefore called the latent heat of steam or latent heat of vaporization. Its quantity varies with the temperature of the steam, and at 212° F. amounts to 970.4 B. T. U. The latent heat at any temperature may be found from the table on properties of saturated steam (pages 252 to 254).

Total Heat of Saturated Steam.—The total heat in one pound of saturated steam may be obtained from the steam table, and consists of three parts:

1. The quantity of heat necessary to bring the water to the temperature of the steam. This is called *sensible heat.*

2. The heat necessary to evaporate the water at the given temperature, which is called *internal latent heat.*

3. The heat used in overcoming the pressure resting on the surface of the water, be it the atmospheric pressure, or steam pressure in the boiler. It performs the work of making room for the steam generated, and is therefore called the *latent heat of volume.*

The volume of one pound of steam at atmospheric pressure is 26.79 cubic feet, and the steam must consequently remove this amount of atmospheric air. The pressure of the atmosphere on one square foot amounts to 14.7×144, or 2,116.8 pounds, and the steam therefore has to raise 2,116.8 pounds to a height of 26.79 feet, or has to perform $2{,}116.8 \times 26.79$, or 56,709 foot-pounds of work.

Since 777.52 foot-pounds of work correspond to one B. T. U., 56,709 foot-pounds are equal to $\frac{56{,}709}{777.52} = 72.9$ B. T. U. This is the latent heat of volume, and deducting these 72.9 B. T. U. from the total latent heat, 970.4, leaves 897.5 B. T. U. as the internal latent heat.

The following formula* gives approximately the total heat in a pound of saturated steam above 32° F.:

$$H = 1{,}150.3 + 0.3745 \times (t - 212) - 0.000550\ (t - 212)^2$$

H = total heat in B. T. U. in steam above 32° F.

t = temperature of steam in degrees Fahrenheit.

Example.—What is the total heat in B. T. U. above 32° F. in one pound of steam of 320° F.?

Solution.—

$$H = 1{,}150.3 + 0.3745\ (320 - 212) - 0.000550\ (320 - 212)^2$$
$$= 1{,}150.3 + 0.3745 \times 108 - 0.00055 \times 108 \times 108$$
$$= 1{,}150.3 + 40.4 - 6.4 = 1{,}184.3 \text{ B. T. U.}$$

Volume of Saturated Steam.—The volume of saturated steam in cubic feet per pound increases in the same ratio as the density decreases, and vice versa, or, in other words, the volume is the reciprocal of the density. Its value at different pressures is given in the table of saturated steam.

* Marks & Davis.

Properties of Saturated Steam.*

Gauge Press. lbs. per Square Inch.	Absolute Press. lbs. per Square Inch.	Temperature Degrees F.	Heat of Liquid above 32 Degrees F. B. T. U.	Latent Heat of Evap. in B. T. U.	Total Heat above 32 Degrees F. in B. T. U.	Volume of 1 lb. in cubic feet.	Weight of 1 cubic foot in lbs.
27.88	1	101.83	69.8	1034.6	1104.4	333.0	0.00300
25.85	2	126.15	94.0	1021.0	1115.0	173.5	0.00576
23.81	3	141.52	109.4	1012.3	1121.6	118.5	0.00845
21.78	4	153.01	120.9	1005.7	1126.5	90.5	0.01107
19.74	5	162.28	130.1	1000.3	1130.5	73.33	0.01364
17.70	6	170.06	137.9	995.8	1133.7	61.89	0.01616
15.67	7	176.85	144.7	991.8	1136.5	53.56	0.01867
13.63	8	182.86	150.8	988.2	1139.0	47.27	0.02115
11.60	9	188.27	156.2	985.0	1141.1	42.36	0.02361
9.56	10	193.22	161.1	982.0	1143.1	38.38	0.02606
7.52	11	197.75	165.7	979.2	1144.9	35.10	0.02849
5.49	12	201.96	169.9	976.6	1146.5	32.36	0.03090
3.45	13	205.87	173.8	974.2	1148.0	30.03	0.03330
1.42	14	209.55	177.5	971.9	1149.4	28.02	0.03569
lbs.gauge							
.3	15	213.0	181.0	969.7	1150.7	26.27	0.03806
1.3	16	216.3	184.4	967.6	1152.0	24.79	0.04042
2.3	17	219.4	187.5	965.6	1153.1	23.38	0.04277
3.3	18	222.4	190.5	963.7	1154.2	22.16	00.4512
4.3	19	225.2	193.4	961.8	1155.2	21.07	0.04746
5.3	20	228.0	196.1	960.0	1156.2	20.08	0.04980
6.3	21	230.6	198.8	958.3	1157.1	19.18	0.05213
7.3	22	233.1	201.3	956.7	1158.0	18.37	0.05445
8.3	23	235.5	203.8	955.1	1158.8	17.62	0.05676
9.3	24	237.8	206.1	953.5	1159.6	16.93	0.05907
10.3	25	240.1	208.4	952.0	1160.4	16.30	0.0614
11.3	26	242.2	210.6	950.6	1161.2	15.72	0.0636
12.3	27	244.4	212.7	949.2	1161.9	15.18	0.0659
13.3	28	246.4	214.8	947.8	1162.6	14.67	0.0682
14.3	29	248.4	216.8	946.4	1163.2	14.19	0.0705
15.3	30	250.3	218.8	945.1	1163.9	13.74	0.0728
16.3	31	252.2	220.7	943.8	1164.5	13.32	0.0751
17.3	32	254.1	222.6	942.5	1165.1	12.93	0.0773
18.3	33	255.8	224.4	941.3	1165.7	12.57	0.0795
19.3	34	257.6	226.2	940.1	1166.3	12.22	0.0818
20.3	35	259.3	227.9	938.9	1166.8	11.89	0.0841
21.3	36	261.0	229.6	937.7	1167.3	11.58	0.0863
22.3	37	262.6	231.3	936.6	1167.8	11.29	0.0886
23.3	38	264.2	232.9	935.5	1168.4	11.01	0.0908
24.3	39	265.8	234.5	934.4	1168.9	10.74	0.0931
25.3	40	267.3	236.1	933.3	1169.4	10.49	0.0953
26.3	41	268.7	237.6	932.2	1169.8	10.25	0.0976
27.3	42	270.2	239.1	931.2	1170.3	10.02	0.0998
28.3	43	271.7	240.5	930.2	1170.7	9.80	0.1020
29.3	44	273.1	242.0	929.2	1171.2	9.59	0.1043
30.3	45	274.5	243.4	928.2	1171.6	9.39	0.1065
31.3	46	275.8	244.8	927.2	1172.0	9.20	0.1087
32.3	47	277.2	246.1	926.3	1172.4	9.02	0.1109

* Marks & Davis' Tables, by permission of Longmans Green & Co.

Properties of Saturated Steam.—Continued.

Gauge Press. lbs. per Square Inch.	Absolute Press. lbs. per Square Inch.	Temperature Degrees F.	Heat of Liquid above 32 Degrees F. B. T. U.	Latent Heat of Evap. in B. T. U.	Total Heat above 32 Degrees F. in B. T. U.	Volume of 1 lb. in cubic feet.	Weight of 1 cubic foot in lbs.
33.3	48	278.5	247.5	925.3	1172.8	8.84	0.1131
34.3	49	279.8	248.8	924.4	1173.2	8.67	0.1153
35.3	50	281.0	250.1	923.5	1173.6	8.51	0.1175
36.3	51	282.3	251.4	922.6	1174.0	8.35	0.1197
37.3	52	283.5	252.6	921.7	1174.3	8.20	0.1219
38.3	53	284.7	253.9	920.8	1174.7	8.05	0.1241
39.3	54	285.9	255.1	919.9	1175.0	7.91	0.1263
40.3	55	287.1	256.3	919.0	1175.4	7.78	0.1285
41.3	56	288.2	257.5	918.2	1175.7	7.65	0.1307
42.3	57	289.4	258.7	917.4	1176.0	7.52	0.1329
43.3	58	290.5	259.8	916.5	1176.4	7.40	0.1350
44.3	59	291.6	261.0	915.7	1176.7	7.28	0.1372
45.3	60	292.7	262.1	914.9	1177.0	7.17	0.1394
46.3	61	293.8	263.2	914.1	1177.3	7.06	0.1416
47.3	62	294.9	264.3	913.3	1177.6	6.95	0.1438
48.3	63	295.9	265.4	912.5	1177.9	6.85	0.1460
49.3	64	297.0	266.4	911.8	1178.2	6.75	0.1482
50.3	65	298.0	267.5	911.0	1178.5	6.65	0.1503
51.3	66	299.0	268.5	910.2	1178.8	6.56	0.1525
52.3	67	300.0	269.6	909.5	1179.0	6.47	0.1547
53.3	68	301.0	270.6	908.7	1179.3	6.38	0.1569
54.3	69	302.0	271.6	908.0	1179.6	6.29	0.1590
55.3	70	302.9	272.6	907.2	1179.8	6.20	0.1612
56.3	71	303.9	273.6	906.5	1180.1	6.12	0.1634
57.3	72	304.8	274.5	905.8	1180.4	6.04	0.1656
58.3	73	305.8	275.5	905.1	1180.6	5.96	0.1678
59.3	74	306.7	276.5	904.4	1180.9	5.89	0.1699
60.3	75	307.6	277.4	903.7	1181.1	5.81	0.1721
61.3	76	308.5	278.3	903.0	1181.4	5.74	0.1743
62.3	77	309.4	279.3	902.3	1181.6	5.67	0.1764
63.3	78	310.3	280.2	901.7	1181.8	5.60	0.1786
64.3	79	311.2	281.1	901.0	1182.1	5.54	0.1808
65.3	80	312.0	282.0	900.3	1182.3	5.47	0.1829
66.3	81	312.9	282.9	899.7	1182.5	5.41	0.1851
67.3	82	313.8	283.8	899.0	1182.8	5.34	0.1873
68.3	83	314.6	284.6	898.4	1183.0	5.28	0.1894
69.3	84	315.4	285.5	897.7	1183.2	5.22	0.1915
70.3	85	316.3	286.3	897.1	1183.4	5.16	0.1937
71.3	86	317.1	287.2	896.4	1183.6	5.10	0.1959
72.3	87	317.9	288.0	895.8	1183.8	5.05	0.1980
73.3	88	318.7	288.9	895.2	1184.0	5.00	0.2001
74.3	89	319.5	289.7	894.6	1184.2	4.94	0.2023
75.3	90	320.3	290.5	893.9	1184.4	4.89	0.2044
76.3	91	321.1	291.3	893.3	1184.6	4.84	0.2065
77.3	92	321.8	292.1	892.7	1184.8	4.79	0.2087
78.3	93	322.6	292.9	892.1	1185.0	4.74	0.2109
79.3	94	323.4	293.7	891.5	1185.2	4.69	0.2130
80.3	95	324.1	294.5	890.9	1185.4	4.65	0.2151
81.3	96	324.9	295.3	890.3	1185.6	4.60	0.2172
82.3	97	325.6	296.1	889.7	1185.8	4.56	0.2193

Properties of Saturated Steam.—Continued.

Gauge Press. lbs. per Square Inch.	Absolute Press. lbs. per Square Inch.	Temperature Degrees F.	Heat of Liquid above 32 Degrees F. B. T. U.	Latent Heat of Evap. in B. T. U.	Total Heat above 32 Degrees F. in B. T. U.	Volume of 1 lb. in cubic feet.	Weight of 1 cubic foot in lbs.
83.3	98	326.4	296.8	889.2	1186.0	4.51	0.2215
84.3	99	327.1	297.6	888.6	1186.2	4.47	0.2237
85.3	100	327.8	298.3	888.0	1186.3	4.429	0.2258
87.3	102	329.3	299.8	886.9	1186.7	4.347	0.2300
89.3	104	330.7	301.3	885.8	1187.0	4.268	0.2343
91.3	106	332.0	302.7	884.7	1187.4	4.192	0.2386
93.3	108	333.4	304.1	883.6	1187.7	4.118	0.2429
95.3	110	334.8	305.5	882.5	1188.0	4.047	0.2472
97.3	112	336.1	306.9	881.4	1188.4	3.978	0.2514
99.3	114	337.4	308.3	880.4	1188.7	3.912	0.2556
101.3	116	338.7	309.6	879.3	1189.0	3.848	0.2599
103.3	118	340.0	311.0	878.3	1189.3	3.786	0.2641
105.3	120	341.3	312.3	877.2	1189.6	3.726	0.2683
107.3	122	342.5	313.6	876.2	1189.8	3.668	0.2726
109.3	124	343.8	314.9	875.2	1190.1	3.611	0.2769
111.3	126	345.0	316.2	874.2	1190.4	3.556	0.2812
113.3	128	346.2	317.4	873.3	1190.7	3.504	0.2854
115.3	130	347.4	318.6	872.3	1191.0	3.452	0.2897
120.3	135	350.3	321.7	869.9	1191.6	3.331	0.3002
125.3	140	353.1	324.6	867.6	1192.2	3.219	0.3107
130.3	145	355.8	327.4	865.4	1192.8	3.112	0.3213
135.3	150	358.5	330.2	863.2	1193.4	3.012	0.3320
140.3	155	361.0	332.9	861.0	1194.0	2.920	0.3425
145.3	160	363.6	335.6	858.8	1194.5	2.834	0.3529
150.3	165	366.0	338.2	856.8	1195.0	2.753	0.3633
155.3	170	368.5	340.7	854.7	1195.4	2.675	0.3738
160.3	175	370.8	343.2	852.7	1195.9	2.602	0.3843
165.3	180	373.1	345.6	850.8	1196.4	2.533	0.3948
170.3	185	375.4	348.0	848.8	1196.8	2.468	0.4052
175.3	190	377.6	350.4	846.9	1197.3	2.406	0.4157
180.3	195	379.8	352.7	845.0	1197.7	2.346	0.4262
185.3	200	381.9	354.9	843.2	1198.1	2.290	0.437
190.3	205	384.0	357.1	841.4	1198.5	2.237	0.443
195.3	210	386.0	359.2	839.6	1198.8	2.187	0.457
200.3	215	388.0	361.4	837.9	1199.2	2.138	0.468
205.3	220	389.9	363.4	836.2	1199.6	2.091	0.478
210.3	225	391.9	365.5	834.4	1199.9	2.046	0.489
215.3	230	393.8	367.5	832.8	1200.2	2.004	0.499
220.3	235	395.6	369.4	831.1	1200.6	1.964	0.509
225.3	240	397.4	371.4	829.5	1200.9	1.924	0.520
230.3	245	399.3	373.3	827.9	1201.2	1.887	0.530
235.3	250	401.1	375.2	826.3	1201.5	1.850	0.541
245.3	260	404.5	378.9	823.1	1202.1	1.782	0.561
255.3	270	407.9	382.5	820.1	1202.6	1.718	0.582
265.3	280	411.2	386.0	817.1	1203.1	1.658	0.603
275.3	290	414.4	389.4	814.2	1203.6	1.602	0.624
285.3	300	417.5	392.7	811.3	1204.1	1.551	0.645
335.3	350	431.9	408.2	797.8	1206.1	1.334	0.750
385.3	400	444.8	422.0	786.0	1208.0	1.17	0.86
435.3	450	456.5	435.0	774.0	1209.0	1.04	0.96
485.3	500	467.3	448.0	762.0	1210.0	0.93	1.08

Properties of Saturated Steam (Vacuum Table*).

Vacuum in Inches of Mercury Referred to 30-inch Bar. (Mercury at 58.4° F.)	Temperature Degrees F.	Volume in Cubic Feet per Pound.	Heat of Liquid in B. T. U.	Total Heat of Steam in B. T. U.	Entropy of Water.	Entropy of Steam.
29.8	34.42	3004.	2.43	**1074.4**	0.0050	2.1748
29.7	44.91	2040.	12.97	1079.2	0.0260	2.1392
29.6	52.60	1554.	20.68	1082.5	0.0412	2.1145
29.5	58.77	1259.	26.85	1085.3	0.0532	2.0951
29.4	63.86	1063.	31.93	1087.5	0.0628	2.0795
29.3	68.33	918.	36.40	1089.6	0.0713	2.0662
29.2	72.27	810.	40.32	1091.3	0.0788	2.0548
29.1	75.84	724.	43.88	1092.9	0.0855	2.0447
29.0	79.07	657.	47.11	1094.3	0.0913	2.0358
28.9	81.97	599.3	50.00	1095.6	0.0968	2.0275
28.8	84.61	552.5	52.63	1096.8	0.1016	2.0203
28.7	87.10	512.2	55.11	1097.9	0.1060	2.0139
28.6	89.47	476.9	57.47	1098.9	0.1104	2.0062
28.5	91.70	446.2	59.70	1100.0	0.1146	2.0015
28.4	93.79	419.6	61.78	1100.9	0.1183	1.9960
28.3	95.78	396.0	63.77	1101.7	0.1219	1.9908
28.2	97.67	375.0	65.65	1102.6	0.1253	1.9860
28.1	99.45	356.4	67.42	1103.4	0.1286	1.9813
28.0	101.15	339.6	69.12	1104.1	0.1316	1.9772
27.9	102.79	324.1	70.75	1104.8	0.1343	1.9729
27.8	104.35	310.3	72.31	1105.5	0.1371	1.9691
27.7	105.85	297.6	73.80	1106.1	0.1398	1.9655
27.6	107.30	286.0	75.25	1106.8	0.1423	1.9620
27.5	108.70	275.2	76.64	1107.4	0.1449	1.9585
27.4	110.05	265.1	77.99	1108.0	0.1472	1.9552
27.3	111.36	255.8	79.30	1108.6	0.1495	1.9521
27.2	112.63	247.2	80.56	1109.1	0.1517	1.9491
27.1	113.87	239.2	81.80	1109.6	0.1539	1.9461
27.0	115.06	231.9	82.98	1110.2	0.1560	1.9434
26.9	116.20	224.6	84.12	1110.7	0.1579	1.9407
26.8	117.32	218.0	85.14	1111.2	0.1599	1.9382
26.7	118.42	211.7	86.33	1111.7	0.1618	1.9356
26.6	119.50	205.8	87.41	1112.2	0.1636	1.9331
26.5	120.55	200.2	88.46	1112.6	0.1654	1.9306
26.4	121.55	195.1	89.46	1113.0	0.1671	1.9283
26.3	122.54	190.1	90.44	1113.4	0.1688	1.9262
26.2	123.51	185.5	91.41	1113.9	0.1704	1.9240

* By permission of Wheeler Condenser & Engineering Co., Carteret, N. J.

Properties of Saturated Steam (Vacuum Table).

(Continued)

Vacuum in Inches of Mercury Referred to 30-inch Bar. (Mercury at 58.4° F.)	Temperature Degrees F.	Volume in Cubic Feet per Pound.	Heat of Liquid in B. T. U.	Total Heat of Steam in B. T. U.	Entropy of Water.	Entropy of Steam.
26.1	124.45	181.0	92.35	1114.3	0.1720	1.9218
26.0	125.38	176.7	93.28	1114.7	0.1736	1.9197
25.9	126.28	172.7	94.18	1114.9	0.1752	1.9176
25.8	127.17	168.9	95.06	1115.3	0.1767	1.9157
25.7	128.04	165.1	95.93	1115.7	0.1782	1.9138
25.6	128.90	161.5	96.79	1116.1	0.1797	1.9119
25.5	129.75	158.1	97.64	1116.5	0.1812	1.9100
25.4	130.59	154.8	98.48	1116.9	0.1826	1.9082
25.3	131.42	151.6	99.30	1117.3	0.1840	1.9064
25.2	132.21	148.6	100.11	1117.7	0.1853	1.9047
25.1	133.00	145.8	100.88	1118.0	0.1866	1.9030
25.0	133.77	143.0	101.65	1118.3	0.1879	1.9013
24.0	140.64	129.0	108.51	1121.3	0.1994	1.8867
23.0	146.78	104.5	114.64	1123.9	0.2096	1.8739
22.0	152.16	92.3	120.02	1126.2	0.2185	1.8631
21.0	157.00	82.6	124.86	1128.2	0.2263	1.8535
20.0	161.42	74.8	129.28	1130.1	0.2334	1.8449
19.0	165.42	68.5	133.28	1131.8	0.2398	1.8372
18.0	169.14	63.1	137.00	1133.4	0.2457	1.8302
17.0	172.63	58.6	140.50	1134.8	0.2512	1.8238
16.0	175.93	54.6	143.80	1136.1	0.2564	1.8177
15.0	179.03	51.17	146.91	1137.4	0.2612	1.8121
14.0	181.92	49.03	149.80	1138.6	0.2658	1.8070
13.0	184.68	45.55	152.57	1139.7	0.2701	1.8021
12.0	187.31	43.18	155.21	1140.7	0.2742	1.7975
11.0	189.83	41.05	157.73	1141.7	0.2780	1.7932
10.0	192.23	39.13	160.14	1142.7	0.2817	1.7890
9.0	194.52	37.40	162.44	1143.6	0.2853	1.7852
8.0	196.73	35.79	164.68	1144.5	0.2887	1.7815
7.0	198.87	34.33	166.81	1145.4	0.2919	1.7779
6.0	200.94	33.00	168.88	1146.3	0.2951	1.7745
5.0	202.92	31.76	170.89	1147.0	0.2981	1.7712
4.0	204.85	30.62	172.81	1147.6	0.3010	1.7680
3.0	206.71	29.55	174.68	1148.4	0.3038	1.7650
2.0	208.52	28.57	176.50	1149.1	0.3065	1.7621
1.0	210.28	27.66	178.27	1149.7	0.3092	1.7593
0.0	212.00	26.79	180.00	1150.4	0.3118	1.7565

Relative Volume of Saturated Steam.—By comparing the volume of a certain weight of saturated steam with that of the same weight of water of maximum density, we obtain the relative volume of saturated steam. It is found by dividing the volume in cubic feet of one pound of saturated steam at a given pressure by the volume of one pound of water of 39° F., which is $\frac{1}{62.425}$ cubic foot.

Example.—What is the relative volume of saturated steam at 92 pounds absolute pressure?

Solution.—From the steam table we find that the volume of one pound of saturated steam at 92 pounds absolute pressure is 4.79 cubic feet. The relative volume is therefore $4.79 \div \frac{1}{62.425}$ $= 4.79 \times 62.425 = 299.0$.

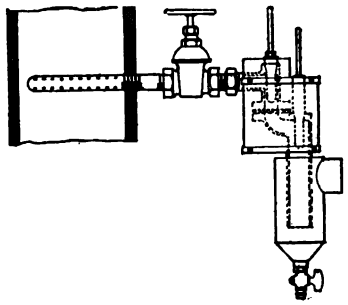


Figure 81. Throttling Calorimeter.

Determination of Moisture in Steam.—Most of the modern forms of fire-tube and water-tube boilers furnish practically dry steam, the latter generally containing not more than 1 to 2 per cent of moisture. Provided the steam contains less than 3% of moisture, the best method for determining the moisture in steam is by means of a throttling calorimeter, this being superior to the separating steam calorimeter, used in former years. Figure 81 shows a *throttling calorimeter.*

The sampling nozzle should be placed in the vertical steam-pipe rising from the boiler. It should be made of ½-inch pipe, and should be so inserted as to extend into or through the steam-pipe to within half an inch of the opposite inside wall of the pipe. It should be closed at the end, and perforated with not less than

twenty ⅛-inch holes equally distributed along and around its cylindrical surface, but none of these holes should be within more than half an inch of the inner side of the pipe. Some engineers prefer to have the nozzle not perforated. The calorimeter, and the pipe leading to it, should be well covered with felting.

Whenever the indications of a throttling or a separating calorimeter show that the percentage of moisture is irregular, or that it is occasionally in excess of 3%, the results should be checked by a steam-separator placed in the steam-pipe as close to the boiler as convenient, with a calorimeter in the steam-pipe, just beyond the outlet from the separator. The drip from the separator should be caught and weighed, and the percentage of moisture computed therefrom, should be added to that shown by the calorimeter.

The formula* for calculating the percentage of moisture when a throttling calorimeter is used, is the following:

$$w = 100 \times \frac{H - h - k \times (T - t)}{L}$$

w = percentage of moisture in the steam.
H = total heat in B. T. U. above 32° F. in one pound of steam.
h = total heat per pound of steam at the pressure in the discharge side of the calorimeter, which is 1,150.4 B. T. U. at atmospheric pressure.
L = latent heat per pound of steam at the pressure in the steam-pipe.
k = specific heat of superheated steam.
T = temperature in degrees F. of the throttled and superheated steam in the calorimeter.
t = temperature in degrees F. at the pressure in the discharge side of the calorimeter, which at atmospheric pressure is 212° F.

Taking k as 0.48 and t as 212°, the formula is reduced to:

$$w = 100 \times \frac{H - 1{,}150.4 - 0.48 \times (T - 212)}{L}$$

Example.—What percentage of moisture does the steam contain if the calorimeter test shows the temperature of the steam entering the calorimeter to be 338° F., and the temperature of the throttled steam in the calorimeter to be 240° F.?

* Wm. Kent, "Mechanical Engineers' Pocket-Book."

Solution.—The steam temperature being 338° F., the total heat H in one pound of such steam is 1,188.8 B. T. U. and the latent heat is 879.9. Consequently we have

$$w = 100 \times \frac{1{,}188.8 - 1{,}150.4 - 0.48 \times (240-212)}{879.9} = 2.86.$$

The percentage of moisture in the steam is 2.86%.

Superheated Steam.

Superheated steam is steam heated to a temperature higher than that due to its pressure at the point of saturation. Only steam not in contact with water can be superheated. For instance, if saturated steam, after it leaves the boiler, is heated by means of hot furnace gases or by direct fire, any moisture in the steam is evaporated, and on further application of heat, the steam becomes superheated.

Pressure of Superheated Steam.—If steam, while being superheated is allowed to expand, its pressure will remain unaltered and only its temperature and its volume will increase. In this case, superheated steam follows closely the laws of perfect gases, the volume increasing in proportion to the absolute temperature.

Specific Heat of Superheated Steam.—The specific heat of superheated steam at constant pressure varies with the temperature and pressure. According to late researches,* the mean specific heat of superheated steam at constant pressure is 0.4762, as compared to the older figure (0.4805) of Regnault, now proven to have been too high (see table on superheated steam, page 265).

Density of Superheated Steam.—The density of superheated steam or the weight in pounds per cubic foot is the reciprocal of the specific volume.

It may therefore readily be found by referring to the table of specific volumes, page 264. The specific volume, at any pressure and any number of degrees superheat, divided into 1 gives the density.

Specific Density of Superheated Steam.—An average figure for the specific density of superheated steam, i. e., the density as compared with air at the same temperature and pressure, may be taken as 0.622. This varies with the temperature and may

* Knoblauch & Jacob, Forschungsarbeiten, Berlin, 1906, Vol. 35 & 36.

be determined at any temperature by means of the following formula:*

$$D = 0.6113 + \frac{0.092 \times t}{850 - t}$$

D = specific density, of superheated steam.
t = temperature in degrees F.

Specific Volume of Superheated Steam.—By specific volume of superheated steam is meant the volume in cubic feet per pound. For specific volumes at various pressures and degrees of superheat, see table, page 264.

Determination of Degree of Superheat.—The amount of superheat should be determined by means of an accurate and reliable thermometer, which is placed in a mercury well, inserted in the steam-pipe. It is the difference between the reading of the thermometer in the steam-pipe and the reading of the same thermometer in saturated steam at the same pressure. The latter must be determined by special experiment, and should not be taken from the steam table.

Advantages of Superheated Steam.—The use of superheated steam offers several important advantages. On account of the extra heat stored in it, it will perform more *work* than saturated steam at the same pressure. If the same amount of heat as is used for superheating were employed to raise the temperature of saturated steam, so as to obtain the same amount of work in both cases, the presssure would be raised entirely too far for ordinary practical purposes.

Furthermore, *condensation,* which entails loss of heat and consequently power, will only take place after the point of saturation has been reached, and superheated steam can therefore be employed to excellent advantage, especially where the steam must be conducted through long pipe lines. Saturated steam will commence to condense at once, the condensed water which covers the inside walls of pipes and steam cylinders causing at the same time a further loss of heat by radiation.

With saturated steam, the loss of heat in a steam engine due to condensation varies from 10 to 50%, whereas by the use of superheated steam, this loss is largely, if not entirely overcome.

For the most efficient results, the amount of superheat should be such that at release the steam will have supplied not only the

* H. M. Murphy, Engineering News, 1908.

heat absorbed by the cylinder walls, but also the heat equivalent of the work done during expansion. Greater superheat than this will result in loss of energy, unless the steam is exhausted into another cylinder.*

Possessing the properties of perfect gases, which are all very poor conductors of heat, superheated steam will lose only very little heat by *radiation*.

The use of a moderate degree of superheat in the average plant will generally insure dry steam in all parts of the piping. If a reasonable superheat is always maintained, it will thus make the use of steam separators unnecessary, for under such conditions, the separator does no useful work and merely offers increased surface for the loss of heat.

In American practice, the employment of moderate superheat has become general, due to the use of Corliss and slide valve engines. These types of engines are not suited to the use of very high superheats, although they are materially benefited by the use of moderate superheat. Generally speaking, the amount of superheat used should be determined by final temperatures and not by any particular degree of superheat. On the average, the final temperature of the superheated steam should not be over 500 degrees F., and in many existing plants where superheaters are added to increase the efficiency of the equipment, a final temperature of 475 degrees Fahrenheit would probably be more advisable. In plants operating on low pressure, this will mean that advantage can be taken of higher superheats than in higher pressure plants. Thus at 100 pounds gauge pressure, saturated steam has a temperature of 338 degrees F., and the addition of 150 degrees of superheat will bring this up to 488 degrees F. If the steam pressure were 175 pounds gauge, with a temperature of 377 degrees F., the addition of only 111 degrees superheat would result in the same final temperature as 150 degrees superheat at 100 pounds gauge pressure. Of course there are exceptions to the above practice, and where poppet- or piston-valve engines are used, higher superheats than those mentioned above can be used, with a corresponding increase in economy.

The use of superheated steam in steam turbines is now standard practice. The saving in this type of prime-mover comes

* Gebhardt "Steam Power Plant Engineering."

principally from a saving in friction losses. Superheated steam is a gas of less density than saturated steam and free from moisture, and hence the fluid friction of entrained moisture on the blades of the steam turbine is eliminated. The effect of moisture in turbine blading is detrimental, as the steam comes in contact with the blades at high velocity, and, with the small clearances found in the turbine, this wearing effect of the entrained moisture affects both the efficiency and life of the turbine.

Another advantage is the fact that superheating increases the *volume*, the pressure remaining the same, the benefit of which may be seen from the example below.

When saturated steam at 100 pounds pressure is superheated 200° F., and taking the specific heat under these conditions at 0.492, the heat required to do this superheating amounts to 200×0.492 or 98.4 B. T. U. The total heat necessary to evaporate one pound of water of 32° F. into steam at 100 pounds gauge pressure is 1,188.8 B. T. U., so that in superheating the steam 200° F., only $\frac{98.4 \times 100}{1,188.8}$ or about 8.3% more heat is necessary than that required to change one pound of water of 32° F. into steam at 100 pounds pressure.

Basing the calculation on a feed-water temperature of 212° F., the heat required to produce a pound of saturated steam at 100 pounds gauge pressure would be 1,008.8 B.T.U. In this case $\frac{98.4 \times 100}{1,008.8}$ or 9.8% of this heat would be sufficient to superheat the steam 200° F.

It has been stated previously that the volume of superheated steam increases with an increase in temperature, and since it is the volume of steam which we utilize in the generation of steam and not its weight, this is of great importance.

The volume of one pound of saturated steam at 100 pounds pressure is 3.886 cubic feet, and by superheating such steam 200° F. we increase its volume to 5.100 cubic feet or by 1.214 cubic feet, which corresponds to a gain of $\frac{1.214 \times 100}{3.886}$ or 31.2% in volume. Assuming a feed-water temperature of 212° F., this means that by expending 9.8% of heat we gain 31.2% in volume, and the net gain by superheating amounts to 31.2 minus 9.8 or 21.4%, which is equal to a saving in fuel of that amount.

Professor *R. H. Thurston,* in a paper on Superheated Steam read before the American Society of Mechanical Engineers, arrives at the following conclusions:

"1. Superheated steam, as hitherto employed in the steam engine, has absolutely no thermodynamic value, that is, it neither raises the upper limit of temperature nor depresses the lower limit.

"2. Superheating has for its sole purpose and result, in the steam engine today, the extinction or reduction of the internal thermal wastes of the engine, consequent upon the phenomenon known as initial or cylinder condensation. Here it is extraordinarily effective, and a small quantity of heat expended in superheating the entering steam effects a comparatively large reduction in the expenditure of steam in the engine.

"3. Superheating is superior to any other known means of reduction of internal waste.

"4. No trouble need now be found at the engine with sufficient superheating under usual conditions of operation, to annihilate cylinder condensation.

"5. The more wasteful the engine, the larger the promise of gain by superheating, and small engines will profit by it more than large, slow engines more than fast, and single engines more than the multiple-cylinder systems.

"6. The larger the waste to be checked in the engine, the farther should the superheating be carried."

The extra expense connected with superheating is not always warranted, and of course in cases where this expense is greater than the economy obtained, it becomes a loss instead of a gain. The comparison of steam engines using superheated and saturated steam should be made on the basis of heat consumed per horsepower per hour.

Some criticism of superheated steam for reciprocating engines is still heard, but wherever difficulties are encountered, the trouble generally results from one of two causes, either the use of excessive superheat or from the use of an improperly designed superheater, which gives fluctuations in superheat. The latter difficulty may be obviated by installing superheater equipment designed so that there is ample metal in the superheater proper for the storage of heat, so as to take care of momentary fluctuations of gas temperatures in the boiler at the superheater locations.

Specific Volume of Superheated Steam.*

Cubic Feet per Pound.

(Knoblauch, Linde and Klebe.)

Absolute Pressure Pounds per Square Inch.	Degrees of Superheat, Fahrenheit.																			
	0	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	175	200	225	250
1	333.0	339.0	345.0	352.0	358.0	364.0	370.0	376.0	381.0	388.0	394.0	400.0	406.0	412.0	418.0	424.0	439.0	454.0	468.0	483.0
5	73.3	74.6	75.9	77.2	78.4	79.7	80.9	82.1	83.3	84.5	85.7	86.9	88.2	89.4	90.6	91.8	94.8	97.8	101.0	104.0
10	38.4	39.0	39.7	40.3	40.9	41.5	42.2	42.8	43.4	44.0	44.6	45.2	45.8	46.4	47.0	47.7	49.2	50.7	52.2	53.7
15	26.3	26.7	27.1	27.6	28.0	28.4	28.8	29.2	29.6	30.0	30.5	30.9	31.3	31.7	32.1	32.5	33.5	34.5	35.5	36.6
20	20.1	20.4	20.7	21.0	21.4	21.7	22.0	22.3	22.6	22.9	23.3	23.6	23.9	24.2	24.5	24.8	25.5	26.3	27.0	27.8
25	16.3	16.6	16.8	17.1	17.3	17.6	17.8	18.1	18.4	18.6	18.8	19.1	19.4	19.6	19.8	20.1	20.7	21.3	21.9	22.6
30	13.7	14.0	14.2	14.4	14.6	14.8	15.0	15.2	15.5	15.7	15.9	16.1	16.3	16.5	16.7	16.9	17.4	18.0	18.5	19.0
35	11.9	12.1	12.3	12.5	12.7	12.9	13.0	13.2	13.4	13.6	13.7	13.9	14.1	14.3	14.5	14.6	15.0	15.5	15.9	16.4
40	10.5	11.2	11.4	11.5	11.7	11.9	12.1	12.2	12.4	12.6	12.7	12.9	13.1	13.2	13.4	13.6	14.0	14.4	14.8	15.2
50	8.5	8.7	8.8	8.9	9.0	9.2	9.3	9.4	9.6	9.7	9.8	9.9	10.1	10.2	10.3	10.5	10.8	11.1	11.4	11.7
75	5.81	5.91	6.00	6.10	6.19	6.28	6.37	6.46	6.55	6.64	6.73	6.82	6.91	7.00	7.09	7.17	7.38	7.60	7.81	8.02
100	4.43	4.51	4.58	4.65	4.72	4.79	4.86	4.93	5.00	5.07	5.14	5.21	5.27	5.34	5.41	5.47	5.63	5.80	5.96	6.12
125	3.58	3.64	3.70	3.76	3.82	3.88	3.94	4.00	4.06	4.11	4.17	4.22	4.28	4.33	4.39	4.45	4.58	4.71	4.84	4.97
150	3.01	3.06	3.11	3.17	3.22	3.27	3.32	3.37	3.42	3.46	3.51	3.56	3.61	3.66	3.70	3.75	3.86	3.97	4.08	4.19
175	2.60	2.65	2.69	2.74	2.78	2.83	2.87	2.91	2.96	3.00	3.04	3.08	3.12	3.16	3.20	3.24	3.34	3.44	3.53	3.63
200	2.29	2.33	2.37	2.41	2.45	2.49	2.53	2.57	2.61	2.64	2.68	2.72	2.76	2.79	2.83	2.86	2.95	3.04	3.12	3.21
225	2.05	2.09	2.12	2.15	2.19	2.23	2.26	2.30	2.33	2.37	2.40	2.43	2.47	2.50	2.53	2.57	2.64	2.72	2.80	2.88
250	1.85	1.88	1.91	1.95	1.98	2.02	2.05	2.08	2.11	2.14	2.19	2.21	2.24	2.27	2.30	2.33	2.40	2.47	2.54	2.61
275	1.69	1.72	1.75	1.78	1.81	1.84	1.87	1.90	1.93	1.96	1.99	2.02	2.04	2.07	2.10	2.13	2.19	2.26	2.32	2.39
300	1.55	1.58	1.60	1.63	1.66	1.69	1.72	1.75	1.78	1.80	1.83	1.86	1.88	1.91	1.94	1.96	2.02	2.09	2.15	2.21

* G. F. Gebhardt "Steam Power Plant Engineering."

Mean Specific Heat of Superheated Steam.*

(Based on Experiments of Knoblauch and Jacob).

Absolute Pressure Pounds per Square Inch.	Degrees of Superheat, Fahrenheit.																			
	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	175	200	225	250	300
1	.452	.452	.453	.453	.454	.454	.454	.455	.455	.455	.455	.455	.455	.456	.456	.456	.456	.456	.456	.457
5	.560	.460	.460	.460	.460	.460	.459	.459	.459	.459	.459	.459	.459	.459	.459	.459	.460	.460	.460	.460
10	.465	.465	.465	.465	.464	.464	.464	.464	.464	.464	.464	.464	.464	.464	.464	.464	.465	.465	.465	.465
15	.470	.470	.470	.470	.470	.470	.469	.469	.469	.469	.469	.469	.469	.469	.469	.468	.468	.468	.468	.468
20	.475	.475	.475	.475	.474	.474	.474	.474	.473	.473	.473	.473	.473	.472	.472	.472	.472	.472	.471	.471
25	.480	.480	.480	.479	.479	.479	.478	.478	.478	.477	.477	.477	.477	.477	.476	.476	.476	.475	.475	.475
30	.485	.485	.484	.484	.484	.483	.483	.483	.483	.482	.482	.481	.481	.481	.480	.480	.479	.479	.478	.477
35	.490	.490	.489	.489	.488	.488	.487	.487	.486	.486	.486	.485	.485	.484	.484	.483	.482	.482	.481	.480
40	.495	.494	.494	.493	.492	.492	.491	.491	.490	.490	.489	.489	.488	.488	.487	.486	.485	.484	.483	.482
50	.509	.508	.507	.506	.504	.503	.501	.500	.499	.498	.497	.496	.496	.495	.494	.493	.491	.490	.489	.487
75	.540	.535	.533	.530	.528	.525	.523	.521	.519	.517	.515	.513	.512	.511	.509	.506	.504	.502	.500	.497
100	.570	.560	.556	.552	.550	.546	.542	.540	.537	.534	.531	.528	.526	.524	.522	.518	.515	.512	.509	.505
125	.590	.585	.580	.575	.570	.565	.560	.556	.552	.548	.545	.542	.539	.537	.534	.532	.524	.520	.517	.510
150	.620	.615	.606	.597	.592	.585	.579	.572	.566	.562	.558	.554	.550	.547	.544	.539	.533	.528	.524	.518
175	.660	.645	.633	.622	.614	.605	.597	.589	.582	.577	.572	.566	.562	.558	.555	.548	.541	.535	.531	.524
200	.690	.675	.657	.645	.634	.623	.614	.605	.596	.590	.584	.578	.572	.568	.563	.556	.548	.542	.537	.529
225	.730	.710	.686	.672	.656	.643	.631	.620	.611	.603	.595	.589	.583	.578	.573	.562	.553	.547	.543	.535
250	.770	.740	.712	.695	.678	.663	.648	.635	.624	.615	.606	.599	.593	.587	.582	.570	.562	.555	.549	.540
275	.800	.770	.746	.722	.700	.681	.664	.650	.638	.629	.620	.610	.602	.596	.590	.578	.569	.561	.555	.545
300	.850	.805	.773	.740	.724	.702	.683	.666	.652	.641	.630	.621	.612	.605	.599	.586	.576	.568	.561	.550

* G. F. Gebhardt "Steam Power Plant Engineering."

Superheaters.

Superheaters may be divided into two classes, those which are installed within the individual boiler setting, utilizing the heat generated by the boiler itself to superheat the steam, and those which are built within a separate setting wih a furnace of their own to generate the heat necessary to superheat all the steam passed through them.

This form of superheater has the advantage of being automatic in its operation and needs no attention other than the occasional cleaning of the outside surface, the same as the boiler heating surface is cleaned, and if properly designed gives an almost constant superheat.

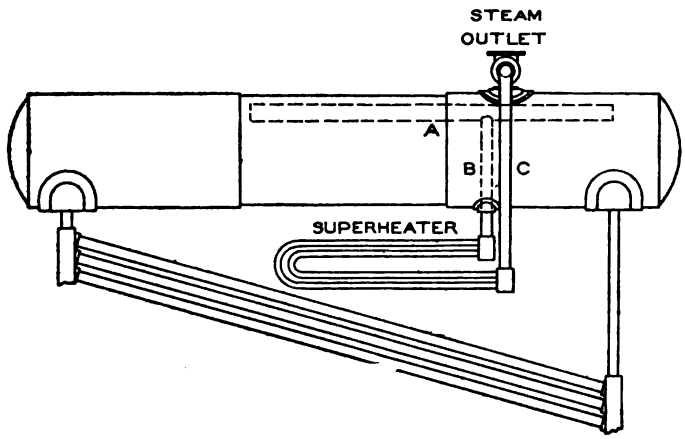


Figure 82. Attached Superheater.

Attached Superheater.—The proper position of a superheater when in connection within the boiler is of the greatest importance. While the temperature to which it is exposed should be high, a certain limit should not be exceeded, and furthermore, the temperature should be as constant as possible. Therefore, it is not good policy to place the superheater too near the furnace or too near the uptake. The farthest end of the combustion chamber is the best location. Figure 82 shows one make of an attached superheater.

Independently-Fired Superheater.—This type of superheater, of which there are several makes on the market, has the advantage of making absolute regulation of superheat possible,

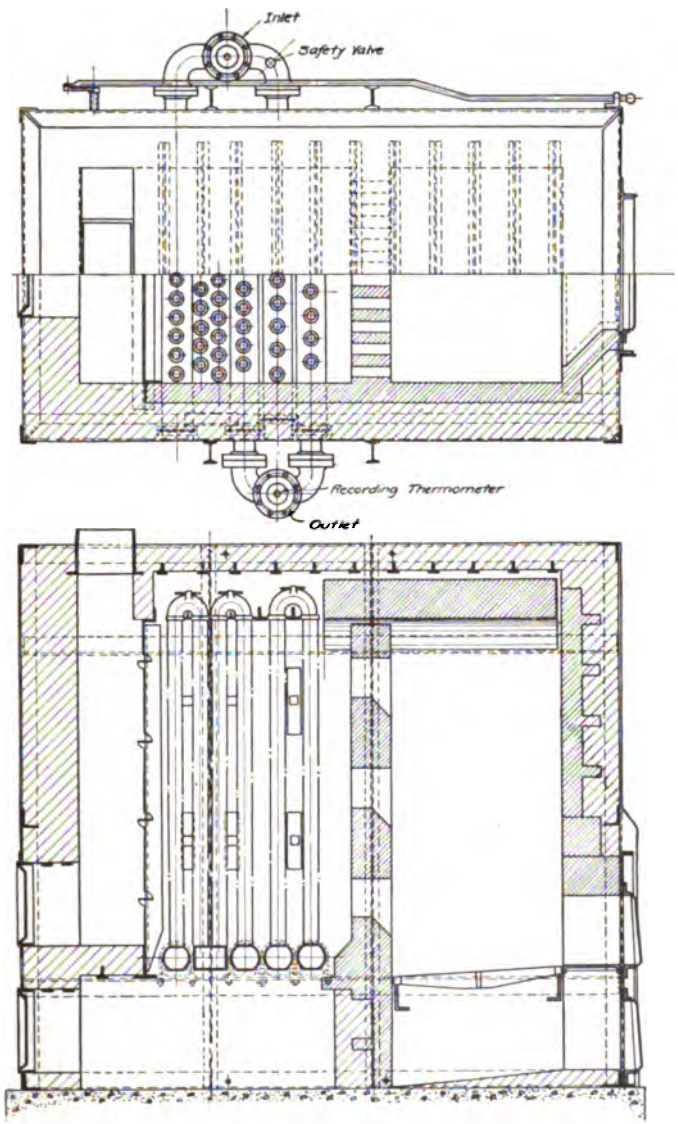


Figure 83. Foster Independently Fired Superheater.

but it requires attention. In large units, these superheaters are built to have an efficiency practically equal to that of the average boiler. In cases where especially high temperatures are required, this form must be used, as the attached superheater will not add more than 200 to 250 degrees of superheat to the steam. Direct fired superheaters have been built to superheat steam as high as 1100° F. final temperature. Figure 83 illustrates the Foster independently-fired superheater.

Flow of Steam.

Flow of Steam through Nozzle.—When steam flows through an opening into any space against an absolute pressure less than 58 per cent of the initial absolute pressure, it has practically a constant velocity, viz., 888 feet per second or 53,280 feet or a little over 10 miles per minute. Consequently, under ordinary conditions, the amount discharged in pounds per minute is proportional to the density of the steam.

To find the theoretical weight of the steam discharged through a nozzle per minute, under ordinary conditions, the following simple formula may consequently be used:

$$W = a \times d \times 370.$$

W = weight of steam in pounds per minute.
a = area of opening in square inches.
d = density of steam in pounds per cubic foot.

This formula is derived as follows:

a = area of pipe opening in square feet.

$\frac{a}{144}$ = area of pipe opening in square feet.

$\frac{a}{144} \times 53{,}280 = a \times 370 =$ cubic feet of steam per minute.

Cubic feet of steam per minute $\times$ density = pounds of steam per minute. Hence, $W = a \times d \times 370$.

Example.—Under ordinary conditions (see above), how many pounds of steam at 100 pounds gauge pressure are theoretically discharged per minute through a pipe 4 inches in diameter?

Solution.—$W = 4 \times 4 \times 0.7854 \times 0.2614 \times 370$
$= 12.566 \times 0.2614 \times 370$
= 1,215 pounds of steam per minute.

By the above calculation, the theoretical flow only is obtained. In practice, less steam is discharged. Rankine has determined that for short pipes the theoretical flow must be multiplied by the factor 0.93, while for calculating the steam flow through an opening in a plate or through a safety valve, the factor 0.63 must be employed.

Referring to the above example, the actual flow through a short pipe would be:

$1{,}215 \times 0.93 = 1{,}130$ pounds of steam per minute.

The discharge through a safety valve would amount to:

$1{,}215 \times 0.63 = 765$ pounds of steam per minute.

Flow of Steam through Pipes.—The rapidity of flow of steam through a pipe depends on the difference in pressure between the inlet and outlet, on the length of pipe, and on the diameter of the pipe.

There are many different formulas for determining the amount of steam flowing through a pipe. The following formula* is the most commonly accepted and gives the approximate weight:

$$W = 87.5 \times \sqrt{\frac{D \times (p_1 - p_2) \times d^5}{L \times \left(1 + \frac{3.6}{d}\right)}}$$

W = weight of steam in pounds per minute.
D = density of steam in pounds per cubic foot on entering pipe.
p_1 = initial gauge pressure in pounds per square inch.
p_2 = gauge pressure in pounds per square inch of steam on leaving pipe.
L = length of pipe in feet.
d = diameter of pipe in inches.

Example.—How many pounds of steam will flow per minute through a pipe 4 inches in diameter and 150 feet long, with a pressure decreasing from 102 to 100 pounds from one end of the pipe to the other, the density at 102 pounds pressure taken from the steam table being 0.2614 pound?

Solution.—

$$W = 87.5 \times \sqrt{\frac{0.2614 \times (102 - 100) \times 4^5}{150 \times \left(1 + \frac{3.6}{4}\right)}}$$

* G. H. Babcock "Steam."

$$= 87.5 \times \sqrt{\frac{0.2614 \times 2 \times 1.024}{150 \times 1.9}}$$

$$= 87.5 \times \sqrt{1.8784}$$

$$= 87.5 \times 1.3705$$

$$= 119.8 \text{ pounds of steam per minute.}$$

The following table gives in approximate figures the weight of steam that will flow in one minute with various initial pressures, and with a loss of pressure of one pound, through straight smooth pipes, having a length 240 times their diameter. The values are fairly close to those obtained by the formula.

Flow of Steam Through Pipes.

Initial Pressure by Gauge. Lbs. Per Sq. In.	Diameter of Pipe in Inches. Length of Each = 240 Diameters.													
	¾	1	1½	2	2½	3	4	5	6	8	10	12	15	18
	Weight of Steam Per Minute in Pounds, With 1 Pound Loss of Pressure.													
1	1.16	2.07	5.7	10.27	15.45	25.38	46.85	77.3	115.9	211.4	341.1	502.4	804	1177
10	1.44	2.57	7.1	12.72	19.15	31.45	58.05	95.8	143.6	262.0	422.7	622.5	996	1458
20	1.70	3.02	8.3	14.94	22.49	36.94	68.20	112.6	168.7	307.8	496.5	731.3	1170	1713
30	1.91	3.40	9.4	16.84	25.35	41.63	76.84	126.9	190.1	346.8	559.5	824.1	1318	1930
40	2.10	3.74	10.3	18.51	27.87	45.77	84.49	139.5	209.0	381.3	615.3	906.0	1450	2124
50	2.27	4.04	11.2	20.01	30.13	49.48	91.34	150.8	226.0	412.2	665.0	979.5	1567	2292
60	2.43	4.32	11.9	21.38	32.19	52.87	97.60	161.1	241.5	440.5	710.6	1046.7	1675	2451
70	2.57	4.58	12.6	22.65	34.10	56.00	103.37	170.7	255.8	466.5	752.7	1108.5	1774	2596
80	2.71	4.82	13.3	23.82	35.87	58.91	108.74	179.5	269.0	490.7	791.7	1166.1	1866	2731
90	2.83	5.04	13.9	24.92	37.52	61.62	113.74	187.8	281.4	513.3	828.1	1219.8	1951	2856
100	2.95	5.25	14.5	25.96	39.07	64.18	118.47	195.6	293.1	534.6	862.6	1270.1	2032	2975
120	3.16	5.63	15.5	27.85	41.93	68.87	127.12	209.9	314.5	573.7	925.6	1363.3	2181	3193
150	3.45	6.14	17.0	30.37	45.72	75.09	138.61	228.8	343.0	625.5	1009.2	1486.5	2378	3481

In case of any *other loss of pressure* than one pound, the correct weight of steam can be found by multiplying the figures in the table by the square root of the given loss. If, for instance, there is a loss in pressure of 4 pounds, we have to multiply the figures in the table by $\sqrt{4}$ or 2.

If the flow of steam is to be calculated for any other *length of pipe,* it must be taken into consideration that the flow decreases as the square root of the increase in length. If, for instance, one pipe is 9 times as long as another, the longer pipe discharges one-third as much steam. Therefore, in the above table, the flow

for any other pipe length can be found by dividing the given length expressed in diameters by 240 and dividing the figures in the table by the square root of the quotient.

Example.—How many pounds of steam of an initial gauge pressure of 100 pounds will flow in one minute through a 2-inch pipe 160 feet long, with a loss of one pound pressure?

Solution.—The length of the pipe is $160 \times 12 = 1{,}920$ inches $= \frac{1920}{2} = 960$ diameters. From the table we find the flow to be 25.96 pounds for a pipe length of 240 diameters. Dividing the flow, 25.96, by $\sqrt{\frac{960}{240}} = \sqrt{4} = 2$, we obtain 12.98 pounds of steam per minute.

Calculation of Horse-Power Delivered through Pipe.—To find the horse-power delivered through the pipe in one hour, multiply the figures in the table by 2. For instance, we find that at an initial gauge pressure of 90 pounds, 24.92 pounds of steam are delivered through a 2-inch pipe; therefore the horse-power delivered would amount to approximately $2 \times 24.92 = 49.84$ H.P.

Decrease of Pressure through Increase of Pipe Length.—The loss of pressure increases in direct proportion to the increase in pipe length, provided the quantity of steam delivered remains the same. For instance, if a 3-inch pipe of a length of 240 diameters delivers 59 pounds of steam per minute with one pound loss of pressure, while another 3-inch pipe of 960 diameters delivers the same weight of steam, the loss in steam pressure will, in the latter case, be four pounds, because the pipe is four times as long.

Loss due to Inertia and Friction.—The loss of head or pressure due to inertia before full velocity is obtained as well as the friction of the steam entering the pipe and passing elbows and valves will reduce the flow as found by the formula, and as given in the preceding table. Such losses are generally allowed for by assuming a length of pipe which has an equivalent resistance, and adding this length to the actual length of pipe.

The resistance at the *opening of a pipe,* and that of a *globe valve* are each about the same as that of a pipe of a length of 114 diameters divided by $1 + \frac{3.6}{\text{diameter}}$.

The resistance of an *elbow* is equal to two-thirds of that of a globe valve.

For the sizes of pipes given in the previous table, the pipe lengths corresponding to opening, globe valves and elbows are:

Pipe Lengths in Diameters Corresponding to Resistance in Openings, Globe Valves and Elbows, for Various Diameters.

Diameter in Inches	¼	1	1¼	2	2½	3	4	5	6	8	10	12	15	18
Openings and Globe Valves	20	25	34	41	47	52	60	66	71	79	84	88	92	95
Elbows	13	17	23	27	31	35	40	44	47	53	56	59	61	63

These equivalents for openings, elbows and globe valves must in each instance be added to the actual length of the pipe. Thus a 4-inch pipe, 40 feet (120 diameters) long with a globe valve and three elbows would be equivalent to: $120 + 60 + 60 + 3 \times 40 = 360$ diameters. At the flow given in the table, the loss of pressure would therefore be $360 \div 240 = 1.5$ pounds.

The quantity of steam delivered with one pound loss would thus be considerably smaller than that given in the table. It can be calculated, as has been shown above, by dividing the flow by $\sqrt{1.5}$. The amount of steam delivered through this pipe would be: $1 \div \sqrt{1.5} = 0.816$, or 81.6% of the weight of the steam indicated in the table.

Calculation of Dimensions of Steam-Pipes for Engines.

Calculation from Steam-Cylinder Area and Piston Speed.— To find the proper size of steam-pipes for stationary engines the following formula, which is based on actual experience, may be used:

$$a = \frac{A \times s}{6{,}000}$$

a = steam-pipe area in square inches.
A = steam-cylinder area in square inches.
s = piston speed in feet per minute.

In this formula, it is assumed that the velocity of the steam does not exceed 6,000 feet per minute.

Table Showing the Amount, etc., of Steam, Leaving a Boiler at 75 Pounds Pressure and Entering Another Vessel at Varying Pressures.

Absolute Pressure in Boiler.	Absolute Exhaust Pressure.	Ratio of Expansion at Nozzle.	Velocity of Outflow at Constant Density, Feet per Second.	Actual Velocity of Outflow, Expanded Steam, Feet per Second.	Discharge per Square Inch Orifice per Minute, Pounds.
75	74.	1.012	227.5	230.	16.68
75	72.	1.037	386.7	401.	28.35
75	70.	1.063	490.0	521.	35.93
75	65.	1.136	660.0	749.	48.38
75	61.62	1.198	736.	876.	53.97
75	60.	1.219	765.	933.	56.12
75	50.	1.434	873.	1252.	64.00
75	45.	1.575	890.	1401.	65.24
75	43.46*	1.624	890.6	1446.5	65.30
75	15.	1.624	890.6	1446.5	65.30
75	0.	1.624	890.6	1446.5	65.30

* Equals 58% of absolute boiler pressure.

Outflow of Steam Into the Atmosphere.

Absolute Initial Pressure per sq. in.	Velocity of Outflow as at Constant Density.	Actual Velocity of Outflow Expanded.	Discharge per sq. in. of Orifice per minute.	Horse-power per sq. in. of Orifice if H. P. = 30 lbs. per hour.
Lbs.	Feet per second.	Feet per second.	Lbs.	H. P.
25.37	863	1401	22.81	45.6
30	867	1408	26.84	53.7
40	874	1419	35.18	70.4
50	880	1429	44.06	88.1
60	885	1437	52.59	105.2
70	889	1444	61.07	122.1
75	891	1447	65.30	130.6
90	895	1454	77.94	155.9
100	898	1459	86.34	172.7
115	902	1466	98.76	197.5

External pressure per square inch 14.7 lbs. Ratio of expansion in nozzle 1.624.

If d represents the diameter of the steam-pipe, the formula becomes:

$$d = \sqrt{\frac{A \times s}{6{,}000 \times 0.7854}}$$

This formula may be simplified so as to read as follows:

$$d = \sqrt{\frac{D^2 \times s}{6{,}000}}$$

D = diameter of the steam-cylinder.

Example.—What should be the diameter of the steam pipe for a steam engine having a cylinder 20 inches in diameter, a stroke of 42 inches and making 60 revolutions per minute?

Solution.—As the piston speed is the stroke times twice the number of revolutions, and as it must be reduced to feet, we obtain:

$$d = \sqrt{\frac{20 \times 20 \times 42 \times 2 \times 60}{6{,}000 \times 12}} = \sqrt{28} = 5.29 \text{ inches.}$$

Calculation from Horse-Power.—If the horse-power of the engine is known, the steam pipe diameter can easily be found by means of the following formula, the results of which tally closely with the findings in actual practice:

$$d = \sqrt{\frac{H.\,P.}{6}}$$

On the other hand, the horse-power delivered can be calculated if the diameter of the steam-pipe is known, as follows:

$$H.\,P. = d^2 \times 6.$$

Example.—What must be the diameter of the steam-pipe for a 160 H. P. engine?

Solution.—

$$d = \sqrt{\frac{160}{6}} = \sqrt{26.67} = 5.16 \text{ inches.}$$

The required diameter is 5.16 inches, and consequently a 5-inch pipe must be used.

Example.—How many horse-power can a 4-inch pipe deliver?

Solution.—$H.\,P. = 4^2 \times 6 = 96.$

Steam Meters.

Where the amount of steam used for different purposes cannot conveniently be determined by weighing the condensed

steam, so-called steam meters may be employed for the purpose. Of these, there are several makes. The Gebhardt steam meter is illustrated in Figure 84. It shows directly the pressure in the line, the velocity of the steam, and the number of pounds per hour.

Figure 84. Steam Meter.

Steam Loops.

The steam loop is a pipe system, by which water of condensation in steam-pipes is automatically returned to the boilers. It usually consists of three pipes, namely the *riser,* the *horizontal* and the *drop-leg.*

The water of condensation in a steam loop is not only required to flow up to the boiler, but must also overcome the difference in pressure. The device must, therefore, perform work, and in so doing, heat will be lost, the heat expended being simply radiation from the upper or horizontal pipe. Figure 85 illustrates the proper arrangement of a steam loop in its simplest form.

Operation of Steam Loop.—Assume the pressure in the boiler to be 100 pounds, at the separator 95 pounds. The latter pressure extends back through the loop, but in the drop-leg meets a column of water which has risen from the boiler, to a height of about 10 feet, i. e., to the hydrostatic head equivalent to the 5 pounds difference in pressure. Thus, the system is placed in

equilibrium. A slight condensation of the steam in the horizontal pipe now occurs, thereby lowering the pressure to about 94 pounds, and the column in the drop-leg rises 6 inches to balance it. Meanwhile, the riser contains a column of mixed vapor, spray and water, which also tends to rise to supply the horizontal as its steam condenses. Being lighter than the water of the drop-leg, it rises much faster. If the contents of the riser has a specific gravity

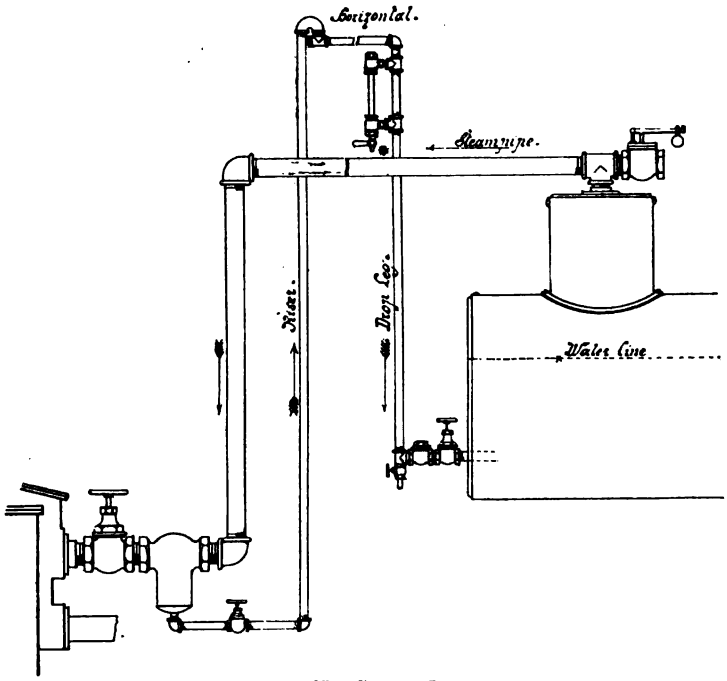


Figure 85. Steam Loop.

of only one-tenth of that of water in the drop-leg the rise will be ten times as rapid, so that when the drop-leg column rises 1 foot, the riser column will be lifted ten feet. By this process the riser will empty its contents into the horizontal pipe, from where it has a free run to the drop-leg and thence to the boiler. No water should accumulate in the separator.

Steam Traps.

A steam trap is a device for draining condensed steam from a steam-pipe or vessel without permitting the escape of steam. The water of condensation accumulates in the trap, and gradually

fills it. When the water has risen to a certain height, a valve, usually placed near the bottom, is opened automatically, by a *float* or some other mechanical means and a certain portion of the condensed water is forced out by the pressure resting upon its surface, until the level of the water is lowered to that point where the float closes the valve again. Steam traps are also constructed, in which a valve is operated by the expansion and contraction of a liquid or metal. These traps are called *expansion traps.*

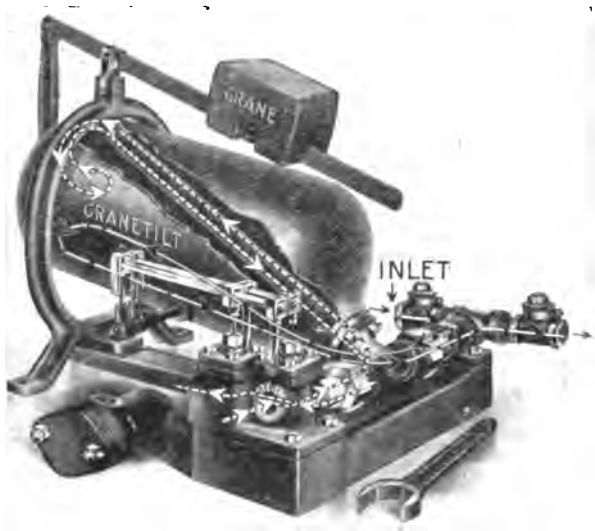


Figure 86. Cranetilt Trap.

So-called *return traps* are traps which return the water of condensation to the boiler having a higher pressure. Such a trap must be located above the water line prevailing in the boiler. *Non-return traps* deliver the water of condensation into a receptacle having a lower pressure.

Figure 86 shows the construction of one type of steam trap.

Steam Separators.

The object of a steam separator is to remove from steam, before it is used in the cylinder of the engine, all liquid water and other liquids or particles of solid matter carried along by the steam from the boiler.

The use of dry steam in an engine cylinder is of the greatest importance, not only for safety, since water present in the cylinder in larger quantities might force out the cylinder head or cause other damage to the engine, but also because the presence of water in the steam considerably reduces the economy of the engine's operation.

The water in wet steam carries away from the boiler a quantity of heat which is incapable of doing work, and increases the initial condensation, one of the principal causes of loss of power in the engine.

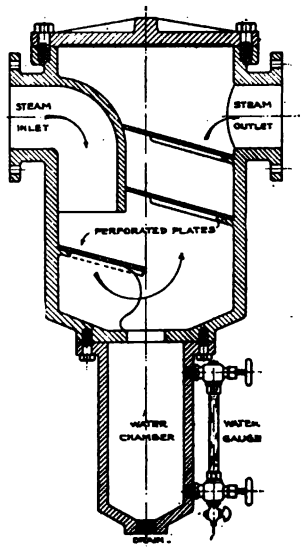


Figure 87. Steam Separator.

Construction and Action of Steam Separators.—The principles upon which the construction and action of steam separators are based are the following:

In the first place, they should be constructed in such a way that the momentum which has been acquired by the suspended liquids and solids is destroyed. This has been accomplished in different manners, for instance, by baffle or deflecting plates, which alter or even reverse the flow of the steam, or by allowing the steam to expand, giving the heavier particles time to fall to the bottom of the apparatus by the action of gravity. After this has been accomplished, it is important to prevent the separated water

from being again taken up and carried along by the purified steam. There are a number of types of proven merit, made for both horizontal and vertical pipes, some of which allow of their being placed directly over the steam cylinder of the engine. Figure 87 illustrates the construction of a steam separator.

Injectors.

The object of an injector is the same as that of a feed-water pump, viz., to force water into the boiler. In the injector, however, the steam comes into direct contact with the water, which has the advantage of heating the water before it enters the boiler. In case, however, there is another apparatus for heating the boiler water, a pump is more economical, since it uses considerably less steam. Nevertheless, boilers are frequently equipped with an injector to be used in case the pump should get out of order. They are also generally employed on portable boilers, such as locomotives, hoisting and threshing machines, etc.

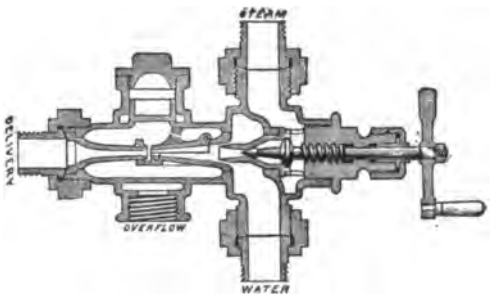


Figure 88. Injector.

Types of Injectors.—We distinguish between *live steam* and *exhaust* injectors, according to the kind of steam employed, the live steam injectors being either *lifting* or *non-lifting.*

A *lifting injector* takes the water from a reservoir standing at a lower level.

A *non-lifting injector* receives its water under pressure either from the city mains or from a tank located at a higher level.

The *exhaust injector* is always non-lifting. It can take the water at 75° F., force it against 75 pounds of pressure and deliver it at about 200° F., these being maximum figures. The lower the temperature of the water received by the injector, the higher can be the lift and the steam pressure.

The lifting and the non-lifting live steam injector may be either *automatic* or *positive.*

The *automatic injector* is so connected with the boiler that a float in the latter either opens or closes the steam valve leading to the injector, so that the proper height of water in the boiler is maintained automatically.

In the *positive injector,* all regulations of steam and water supply are manipulated by one lever and both steam nozzle and water nozzle are fixed.

Operation of Injector.—The possibility of forcing water by means of the boiler pressure against this same pressure is due to the difference in the velocities of steam and water, that of the steam, according to its pressure, being approximately 16 to 18 times greater than that of the water.

The following example shows how an injector forces water into a boiler.

The general formula for the velocity of water flowing from an orifice is:

$$v = \sqrt{2 \times g \times h}.$$

v = velocity of water, in feet per second.
g = gravitation (32.2 feet per second).
h = head or height of water column in feet.

Assuming the steam pressure in a boiler to be 15 pounds per square inch, then, according to the above formula, the velocity of the water flowing from the boiler would be:

$$v = \sqrt{2 \times 32.2 \times \frac{15 \times 34}{14.7}} \text{ or } 47.3 \text{ feet per second.}$$

The steam, however, flows out of the boiler with a velocity of approximately 1,368 feet per second, into the place where it is condensed. If we now assume that the steam strikes the condensing water with this velocity, and that it requires 7 pounds of water to condense 1 pound of steam, then the quantity of motion which the mass of one pound of steam contains is distributed on $7 + 1 = 8$ mass units, and the velocity is consequently reduced to $\frac{1}{8}$ of the original velocity of 1,368 feet per second, or to 171 feet. Assuming friction to cause a loss of 50%, then the final velocity of the feed-water entering the boiler is 171×0.50 or 85.5 feet. Seeing that the water flows from the boiler with a

velocity of only 47.3 feet, it will readily be perceived that the feed-water possesses sufficient velocity to force its way into the boiler against the boiler pressure. The injector works better the lower the temperature of the feed-water, since less mass units must be forced into the boiler.

The loss of work in the injector, due to friction, reappears as heat which is carried into the boiler, and the heat which is converted into useful work in the injector appears in the boiler as stored-up energy. The injector as a boiler feeder has a perfect efficiency; nevertheless it is not the most economical device for the purpose, since it can draw only cold or moderately warm water, while a pump can feed water previously heated by exhaust steam or by the heat of the chimney gases, which would otherwise be wasted.

Duty of Injectors.—Considered as a pump the injector has an exceedingly low efficiency, the duty ranging from 160,000 to 3,300,000 foot-pounds, whereas small pumps such as are used for boiler feeding show a duty of 4,000,000 to 8,000,000 foot-pounds. The duty of an injector can be calculated by means of the following formula:

$$N = \frac{h \times 1{,}000{,}000}{t_1 - t}$$

N = duty in foot-pounds per 1,000,000 B.T.U. furnished by boiler.
h = head of water column corresponding to boiler pressure = $\frac{\text{boiler pressure} \times 34}{14.7}$.
t_1 = final temperature of water in degrees F.
t = initial temperature of water in degrees F.

Example.—What is the duty of an injector in foot-pounds per 1,000,000 B. T. U., if 147 cubic feet of water enter it per hour at a temperature of 66° F. and leave it at 140° F., and if the steam pressure is 100 pounds?

Solution.—

$$N = \frac{100 \times 34 \times 1{,}000{,}000}{14.7 \times (140 - 66)} = 3{,}125{,}500 \text{ foot-pounds.}$$

The duty of the injector is 3,125,500 foot-pounds.

Instead of calculating the head of water corresponding to 100 pounds steam pressure as above $\left(\frac{100 \times 34}{14.7}\right)$, we can also

find the head as follows: $\frac{100 \times 144}{62.3}$, where 144 represents square inches in 1 square foot, and 62.3 is the weight in pounds of one cubic foot of water at 66° F. The latter method of calculating the head of water gives slightly higher and more accurate results.

Calculation of Injector Nozzle.—To find the area of the injector opening necessary to deliver a given volume of water, the following formula may be used:

$$a = \frac{Q}{800 \times \sqrt{P}}$$

a = area of nozzle in square inches.
Q = maximum cubic feet of water per hour.
P = boiler pressure in atmospheres.

Using the figures of the last example, we have:

$$a = \frac{147}{800 \times \sqrt{6.803}} = 0.0705 \text{ square inch.}$$

The diameter of the nozzle will then be: $\sqrt{\frac{0.0705}{0.7854}} = 0.299$ inch, or approximately 19/64 inch.

Boiler Water and its Treatment

In the operation of the modern steam plant, the question of boiler feed-water and its treatment is of vital importance. The most commonly encountered boiler evil, which is to be traced back to the ingredients contained in the boiler feed-water, is scale formation. However, the disturbances caused by corrosion of the boiler iron or by foaming of the water in the boiler, are not to be underestimated.

Water in General.

Chemically pure water does not exist in nature. The nearest approach is rain water or melted snow, but these are never pure, for in falling, the water dissolves or washes down with it, the gases, dust and other particles always present in the air or in the path of its flow. Condensed water, as obtained in the power plant is almost pure, containing only a very slight amount of mineral ingredients, but when originating from the exhaust steam of engines it is liable to contain small quantities of lubricating oil, usually in emulsion.

Rain, after falling on the earth, soaks down into it, and, in percolating through the various strata, dissolves gases, especially carbonic acid gas and oxygen, and certain mineral substances, the quantity and kind of which vary according to the nature of the strata through which the water flows and to the depth to which it passes.

Springs generally are the sources of rivers and the feeders of lakes. By contact with the air, the water flowing from the springs loses part of the mineral ingredients dissolved therein. The water in lakes and streams, furthermore, is diluted by the rain which falls into it and by the admixture of rainwater running off the surrounding land. In this way, such water is made softer, i. e. the contents of mineral substances are decreased. It, however, is enriched by organic matter, consisting of decomposing vege-

table substances, and originating from sewage, which together with other waste water is emptied into the stream or lake by towns or cities situated along or near the water's edge.

Such rainwater as does not run into streams or lakes passes through the earth and collects in underground cavities or flows into subterranean streams. The deeper below the earth's surface such water is taken, the harder it usually is, since the opportunities for dissolving mineral substances are the greater, the deeper the water has sunk into the earth.

Soft and Hard Waters.

As far as scale formation is concerned, boiler feed-waters may be considered from two points of view, namely soft water and hard water.

Soft waters are those that contain very little solids in solution such as can cause the formation of a boiler scale.

Hard waters are such waters that contain considerable or large amounts of mineral substances in solution, particularly calcium and magnesium carbonates and calcium sulphate.

The hardness of a water may be designated as either temporary or permanent.

By **temporary hardness** is meant the hardness caused by such earthy salts as are precipitated when part of the carbonic acid combined with them is removed, for instance, by boiling or aeration. The salts producing temporary hardness in water are calcium, magnesium and iron carbonates, which are dissolved in the water when carbonic acid gas (contained in all waters in nature) is present, but become almost insoluble as soon as this carbonic acid gas is removed. These salts when in solution are in the form of bi-carbonates.

Permanent hardness of a water is caused by the earthy salts remaining in solution, and not precipitated by boiling at 212° F. or by aeration. The chief salt causing it is calcium sulphate.

The best waters for boiler purposes undoubtedly are rain water, melted snow or condensed water, since they contain practically no solids in solution. Next to these come lake and stream or river waters which contain from 5 to 15 grains of solids per gallon, and lastly, well waters. The deeper the source of the latter the harder the water usually is.

Typical American Waters.*

(Laboratory of Wahl-Henius Institute.)

Type of water.	Sodium chloride	Sodium sulphate	Sodium carbonate	Sodium silicates	Calcium sulphate	Calcium carbonate	Magnesium sulphate	Magnesium carbonate	Calcium & magnesium chlorides	Iron sulphate	Free sulphuric acid	Reaction	Total solids
Very Soft Water—Mountain Spring in Allegheny Mountains	trace	..	..	..	trace	1.5	..	0.5	..	..	..	neutral	2.4
Medium Soft Water—Shallow Well, Chicago	0.7	..	..	..	4.4	2.0	..	3.4	..	..	..	neutral	12.8
Medium Hard Water—200 foot Well, Chicago	3.0	trace	..	..	8.8	8.4	..	7.8	..	..	..	neutral	28.4
Medium Hard Water—Deep Well, Chicago	4.8	trace	..	..	5.7	14.8	..	trace	5.4	..	..	neutral	32.2
Lake Superior Water—Duluth	1.0	..	..	..	trace	3.1	..	1.2	..	..	..	neutral	7.4
Lake Michigan Water—Chicago	0.9	..	..	..	2.0	4.8	..	1.5	..	..	..	neutral	10.8
Lake Erie Water—Cleveland	1.3	..	..	..	1.4	3.8	..	1.9	..	..	..	neutral	9.0
Ohio River Water—Filtered†	1.5	..	..	..	3.8	1.3	..	1.1	..	..	..	neutral	9.1
Mississippi River and Missouri River Waters, Filtered—St. Louis	1.3	..	..	..	5.0	trace	..	1.9	..	..	..	neutral	7.9
City Water—San Francisco	3.1	1.4	..	..	0.8	4.0	..	3.1	..	..	..	neutral	13.9
Strongly Alkaline Water—Texas	2.4	0.5	28.5	..	..	trace	..	trace	..	..	..	alkaline	33.8
Mildly Alkaline Water—Various parts of U. S.	2.4	3.1	5.0	..	..	4.2	..	4.2	..	..	..	alkaline	23.2
Water of High Temporary Hardness	4.5	..	..	..	4.0	13.8	..	10.8	..	..	..	neutral	36.1
Water of High Permanent and Temporary Hardness	10.2	..	..	..	28.1	11.5	..	13.6	..	..	..	neutral	69.5
Shallow Well Water—Southern Coast of California	1.0	..	..	4.9	..	1.1	..	2.0	..	..	..	alkaline	11.5
River Water—Originating from Coal Mining District	..	7.2	..	..	15.7	..	4.0	..	..	1.1	2.4	acid	32.9

* Figures represent grains per gallon.

† This water as drawn from the river was very muddy in appearance and contained 38.2 grains per gallon of suspended particles removable by sedimentation and filtration.

On account of drawing off the steam and adding fresh water, large quantities of solid substances, originating from the water, accumulate in the boiler even when a moderately soft water is employed. For example, if a medium soft water containing 10 grains of incrustating solids per gallon is used, and calculating the evaporation as 3½ gallons per horse-power, a 200 H. P. steam boiler, operated 20 hours per day, would allow the accumulation of 10 pounds of dry scale daily, providing none of the sludge or mud in the boiler was removed through the blow-off.

Water for boiler feeding purposes should not be judged by appearance. A good drinking water may be very poorly suited for boiler purposes and cause the formation of a hard scale or corrosion of the boiler iron, while a river water, more or less muddy in appearance, may be well suited on account of its softness, after the particles in suspension that cause the muddiness are removed by sedimentation or filtration. The proper method of determining the suitability of a water for boiler use, is to have an analysis made by a reliable chemist. An opinion as to the quality of the water can then be given and also the best method of treatment in order to minimize the possibility of boiler scale formation, corrosion or foaming.

Boiler Scale.

The formation of boiler scale is the most common evil resulting from the water used for boiler feeding. It nearly always is a more or less hard, stone-like crust, composed chiefly of calcium and magnesium salts which have deposited on the walls and tubes of the boiler and become baked thereon. On account of the great non-conductivity of such a coating, the scale acts as an insulator between the heated metal and the water that is to be heated, and thereby to a certain extent prevents the transfer of the heat, generated from the fuel, to the water. This causes part of the heat to pass into the chimney unused. In consequence, the working, or rather steam generating, capacity, as well as the economical operation, of the boiler will be decreased correspondingly.

Many engineers consider only the loss of heat brought about by scale formation, but this really is not as important as is the danger of overheating the boiler plates and tubes, which ultimately results in a weakening of the boiler iron and may lead finally to a boiler explosion. It does not necessarily follow that the thicker the boiler scale is, the greater will the loss of heat be, because the

nature and composition of the scale are a big factor in this respect, especially if heavy oil or grease, even in small quantities, is present therein.

It is usually conceded that a fair average loss of fuel is to be placed at about 8 per cent for every 1/16 inch in thickness of the scale, but it may be as high as 15 per cent.

Scale of considerable thickness has frequently proven of such insulating power as to allow the iron to become red-hot while keeping up the amount of heat necessary to maintain the required steam pressure. In this red-hot, and consequently soft condition, the plates and tubes, owing to the high internal pressure, are likely to become bent out of shape, viz., to bag, or collapse entirely. There is the further danger that the scale may suddenly crack and break away from the iron, thereby allowing the water to come in contact with the red-hot metal, which often results in an explosion. Heavy scale also increases unequal expansion of the entire boiler structure, which in time causes leaky seams and cracks in the plates near the rivets.

Soft scale or mud in suspension in the boiler water is liable to choke feed pipes, water and steam gauge tubes, promote leaking of cocks and valves and it may be carried over with the steam into the engines.

Scale-Forming Substances.

The scale-producing ingredients of a water may be present either in suspension or in solution, or both. The particles in suspension usually consist of clayey, together with insoluble organic, substances. The dissolved scale-forming salts commonly found in most waters are calcium sulphate, calcium carbonate and magnesium carbonate; sometimes also iron carbonate.

A general description of the individual substances causing scale formation follows:

Calcium Sulphate or **Gypsum** forms a dense, non-porous scale of almost metallic hardness. Its elimination is the first and principal consideration in the treatment of a boiler water. It dissolves more readily in cold than in hot water, and at 290° F. becomes insoluble in water.

Calcium Carbonate is practically insoluble in water at ordinary temperature (about two grains per gallon at 60° F.), and entirely so at 290° F., the temperature at about 50 pounds

boiler pressure. It dissolves when free carbonic acid gas is contained in the water and forms calcium bi-carbonate. When thus dissolved, it begins to precipitate slowly when the water is aerated or heated to about 175° F., and practically all of it becomes insoluble at 212° F. This is due to the expulsion of the loosely bound molecule of carbonic acid gas by the heat.

Calcium carbonate does not settle easily while the boiler is operated, due to its somewhat flocculent condition and to the circulation of the water in the boiler. The true calcium carbonate scale which may be formed in a boiler is comparatively soft and quite porous. It can readily be removed, except when deposited on very hot surfaces not exposed to circulation or when mixed with calcium sulphate, clay, oil or grease.

Magnesium Carbonate is identical in properties to calcium carbonate.

Magnesium Sulphate, although easily soluble in water, hinders the removal of the calcium sulphate and the calcium and magnesium carbonates. It is changed to magnesium carbonate when sodium carbonate (soda ash) is added to the water in the boiler. Magnesium sulphate is not corrosive to iron, nor is it liable to cause foaming of the boiler water.

Calcium Chloride and **Magnesium Chloride** are similar to magnesium sulphate in their action as far as boiler scale formation is concerned, and are eliminated in the form of carbonates by the addition of sodium carbonate.

Magnesium Oxide is never present in water, but is usually a constituent of boiler scale. It is produced when the magnesium carbonate, which has deposited on the plates and tubes, is heated to such an extent that the carbonic acid gas is driven off.

Iron Carbonate acts like calcium and magnesium carbonates, but precipitates at a lower temperature, and especially if the water is exposed to air, viz., aerated. During this decomposition, all of the carbonic acid gas is liberated and the iron takes up oxygen, forming iron oxide (rust).

Silicon and Aluminum Oxides are contained in almost every water, but on account of the very small quantities usually present in solution, are of little importance.

Clay, Sand and **Insoluble Organic Substances** are frequently found in suspension. They increase the bulk of the scale-

producing substances which accumulate in the boiler, intermix with them and facilitate the formation of a hard scale.

Oil and **Grease** entering the boiler, mix with the scale and make it impervious to moisture, thereby increasing its non-conductivity or insulating properties.

Substances Causing Boiler Corrosion.

Besides scale-forming solids, water may contain certain substances that exercise a corrosive or dissolving action on the iron of a boiler. This corrosion is often a greater annoyance than scale, since it may bring about leaks or weakening of the boiler iron to such an extent that costly repairs are needed, which also involves much loss of time by stopping of the boiler while making the repairs.

The corroding substances are generally present in moderate quantities only, but become troublesome when the water is concentrated. Some are readily decomposed by the high temperatures in the boiler, and free acids liberated, which will either attack the iron directly or set up a galvanic action when brass or copper (from cocks, valves, etc.) is in contact with the iron, and as a result the iron of the boiler is slowly dissolved. The substances usually causing corrosion are described below.

Magnesium Chloride.—This salt easily decomposes at boiler temperature, and free hydrochloric acid is produced. The latter not only corrodes the iron of the boiler, but in gaseous form passes over with the steam into the engine, so that further corrosion may result here. Ammonium chloride or sodium chloride (table salt), when present, will to some extent prevent this decomposition, by forming a stable double salt with the magnesium chloride.

Calcium Chloride, when contained in considerable quantities, is claimed by some authorities to act similarly to magnesium chloride. In the presence of magnesium sulphate, the calcium chloride reacts with it and forms magnesium chloride and calcium sulphate.

Decomposing Organic Substances usually must be present in large quantities and be acid in reaction before they have a corrrosive effect.

Free Acids are sometimes found in waters of certain districts. Mine run waters frequently contain free sulphuric or

hydrochloric acid, and these may also be found in river waters into which waste water from steel mills or other factories has been emptied.

Soluble organic acids may get into water when it percolates through beds of decaying organic substances or peat. Water may, furthermore, be badly contaminated by decomposing animal or vegetable substances of acid reaction.

Gases, such as carbonic acid gas, air (oxygen) and sulphuretted hydrogen are corrosive when in a moist state and will readily attack iron. Distilled water free from air or other gases will not corrode or rust iron, nor will dry air or carbonic acid gas. In the boiler, these gases will either form bubbles and cause what is called pitting, or if present in large quantities, which sometimes is the case when air is forced in on account of a defective boiler feed pump, and being heavier than steam, will form a stratum between the water and the steam, this resulting in a corrosion of the iron at the water line.

Substances Causing Foaming of the Boiler Water.

Foaming, priming or frothing of the water in the boilers has no deleterious effect on the boiler iron, but causes the water and any particles suspended therein to be carried over with the steam into the pipes, valves and engines. The most frequent cause of foaming is a too concentrated condition of the boiler water, due to excessive amounts of soluble substances, especially salts, such as sodium chloride and sodium carbonate, being present. Unless the water used for boiler feeding is alkaline, the presence of too much sodium carbonate is to be attributed to the addition of excessive amounts of a boiler compound or water-softening chemical at a time, of which sodium carbonate is a constituent.

Large quantities of soft scale or mud from the feed-water also produce foaming, since the steam bubbles pass through the water in the boiler irregularly and in large bursts or bumps, carrying up a spray of water and mud into the steam pipes.

Foaming usually occurs most easily when very much steam is drawn from the boilers very suddenly and thereby the steam pressure rapidly lowered. If the steam, containing very much water and mud, passes into the steam cylinder of an engine, serious damage frequently results, usually the cylinder head being forced off.

Treatment of Boiler Water.

The treatment of a boiler water, in order to minimize scale formation and prevent corrosion, has long been an interesting study to both the chemist and the practical man. A chemical analysis of the feed-water used in a boiler plant is absolutely necessary nowadays, because only when the constituents of the water and the amounts thereof are known can the best methods for neutralizing or purifying it be adopted. Boiler compounds have been known for years and are still employed most commonly as a boiler water corrective. In late years, however, the complete softening of water before it enters the boiler has been making much headway, and especially when using a hard water, this method of treating the boiler water is to be greatly preferred.

The purification or softening of a water for boiler feeding may be carried out in several different ways. In order to remove the detrimental substances as completely as possible, it is, in most instances, best to treat the water according to several methods. These methods of purification may be outlined as follows:

1) *Mechanically*
   a) By filtration or sedimentation of a muddy water.
   b) By blowing off water from the boiler, thereby flushing out precipitated solids.
2) *Thermally*
   a) By passing water through a feed-water heater.
   b) By mixing the feed-water with condensed water.
3) *Chemically*
   a) By adding a so-called boiler compound to the water in the boiler.
   b) By treating the feed-water with suitable chemicals in an external softening apparatus.

Mechanical Purification.—Mechanical purification by filtration or sedimentation is necessary when the water is in a muddy condition. When water contains very much suspended particles, it is preferable to collect it in suitable tanks or a large reservoir and allow it to stand for 12 to 24 hours, during which time the particles settle. The water, if necessary, can then be filtered. If special tanks are available for the boiler feed-water, it is advisable to treat the water chemically when filling them, so as to precipitate

the calcium and magnesium salts as completely as possible from the cold water. In that case, the treated water should preferably be passed through the feed-water heater first and then through the filter before it enters the boiler, since the heating of the water causes more of the scale-forming substances to become insoluble.

Blow-off valves, with which every boiler is equipped, also allow a mechanical purification of the water in the boiler. The upper blow-off valve performs the function of a skimmer, so that any scum, floating on the surface of the water, may be removed. The lower blow-off valve is situated at the lowest point of the boiler, and while the latter is in operation, much of the precipitated

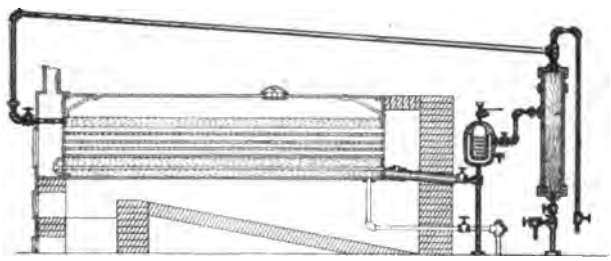


Figure 89. Reiter Mechanical Boiler Cleaner.

solids settle and accumulate near this blow-off valve. If a small quantity of the boiler water is forcibly ejected through this valve, a considerable amount of the mud or sludge is carried along, and in this way the boiler water is purified. The mistake frequently made in many power plants is that the blow-off valves are not used sufficiently often. Good results usually are obtained by blowing off a small quantity of the boiler water every 4 to 6 hours, and the more precipitated solids or soluble salts are contained in the boiler water, the oftener should the blow-off valve be used.

The removal of precipitated solids from the boiler water may also be accomplished by means of *mechanical devices,* in which the water is taken from the upper strata in the boiler and automatically circulated into a settling tank or drum, through which the water passes so slowly that the solids gradually settle, and the almost clear water then flows back into the boiler. A well known and simple device of this kind is the "Reiter Boiler Cleaner" which is illustrated by Figure 89. There are other devices similar in

principle of operation by means of which the muddy water is slowly taken out of the boiler, the precipitated substances removed by sedimentation or filtration and the water then returned to the boiler.

Thermal Softening of Water.—The removal of at least a portion of the scale-forming solids from the boiler feed-water is accomplished by heating the water either while it is in the feed-water supply tank or when it passes through the feed-water heater. The nearer the temperature of the heating approaches 212° F., the more will those scale-forming substances precipitated by the heating, namely the calcium, magnesium, and iron bi-carbonates be made insoluble, since the excess molecule of carbonic acid gas will be expelled, and the resulting carbonates are insoluble, excepting about 2 grains per gallon.

If a special feed-water supply tank is available, a water containing considerable quantities of calcium, magnesium and iron carbonates can readily be softened by heating the water to a temperature of 175° F. or more, or by boiling. It should then be allowed to stand quietly so that the precipitated solids settle. Only the clear water above the sediment should be drawn off. The heated water can also be filtered through a small filter, in order to remove the precipitated particles. Usually when a special tank of this kind is employed, the water, if necessary, is treated at the time of heating, with chemicals in order to remove other detrimental soluble salts which could produce boiler scale or cause corrosion of the boiler iron.

Another, and a further advanced form of thermal purification of the boiler water, is by means of the so-called *live-steam feed-water purifiers*. These devices allow a heating of the feed-water under boiler pressure but are only properly effective when the boiler pressure is higher than 50 pounds, and at such high temperatures, the calcium, magnesium and iron carbonates and the calcium sulphate are completely precipitated from the water. The latter runs over shallow pans, and the scale-producing salts of the water collect on the top as well as on the under side of the pans. The resulting feed-water when it enters the boiler is non-incrustating, and unless it contains substances which corrode iron, does not need the addition of any other chemicals. Should

the feed-water contain such, the addition of soda ash or caustic soda in the necessary quantity should be made before the water enters the live steam purifier.

Under thermal purification, the utilization of the exhaust steam and condensed water also must be included. No modern power plant nowadays allows either of these to go to waste, but employs the exhaust steam for heating the feed-water and for other heating purposes, and the resulting condensed water is then mixed with the water pumped into the boiler. Such condensed water is free from incrustating or corroding substances. Besides this, the condensed water usually has a temperature of about 200° F. or more. The economy of employing the condensed water and exhaust steam can therefore easily be understood. The utmost care, however, must be taken so that the condensed water as well as the exhaust steam when it comes directly in contact with the feed-water, is freed from oil, since should the latter enter into the boiler serious results may follow. The removal of the oil can easily be accomplished by means of devices called oil separators or oil eliminators.

Chemical Purification of Boiler Water.—Filtration or sedimentation and ordinary heating of the feed-water, very often are beneficial towards eliminating a part of the scale-forming substances, but such treatment will not remove the most dangerous scale-producer in most waters, namely the calcium sulphate, nor the substances causing the corrosion of the boiler iron. For neutralizing the corroding substances, it is absolutely necessary to add certain chemicals, usually alkalies, that change them to soluble or insoluble non-corroding salts. The removal of the calcium sulphate can be accomplished by adding a sufficient quantity of any one of a large number of chemicals that transform it into insoluble calcium salts, light and flocculent, so that the newly formed salt can be removed by sedimentation or filtration, if the water was treated before it entered the boiler; or should the chemical first have been added to the water in the boiler, the thus precipitated solids can be expelled in part by allowing small quantities of the boiler water to flow out through the blow-off valve.

In order to avoid the formation of a very hard scale, not alone by the calcium sulphate but also any other scale-producing ingredients, certain substances sometimes are used, which when

added to the water in the steam boiler, will prevent the precipitated calcium sulphate crystals, or any others from uniting into a hard scale. Such substances do not effect a chemical change, but have merely a mechanical action. Their employment, however, still is to be treated under the heading of chemical purification of water.

As already stated, the chemical purification of boiler feed-water may be divided into two main classes, consisting of:

a) Addition of a so-called boiler compound to the water in the boiler.

b) Treatment of the feed-water with suitable chemicals in an external softening apparatus.

Boiler Compounds.

By the term *boiler compound* is generally understood a substance which is added to the water immediately before or after entering the boiler, with the object of neutralizing the corroding ingredients, and affecting, either chemically or mechanically, those incrustating salts which are liable to cause the formation of a hard, rock-like boiler scale, so that they either will produce a soft scale or form a mud or sludge.

In former years there were innumerable boiler compounds sold, and the greatest secrecy was maintained as to their composition. Few of them were of real merit. Many consisted of 90 to 99% of water. The weight of others was wilfully increased by the addition of inert substances, usually ordinary table salt. Some, while preventing scale formation, brought about corrosion of the boiler iron. Exorbitant prices frequently were asked and obtained by the manufacturer of some boiler compound, when the credulous purchaser listened to and accepted the wonderful claims of the merits of the article offered, the composition of which was a great secret, according to the salesman. A chemical analysis usually showed that the compound contained sodium carbonate (soda ash) as the principal scale-preventing ingredient, and the remaining constituents were either of little or no value for the desired purposes.

With the application of scientific principles to the problem of treating boiler waters, it was soon realized that every individual type of water needs special treatment. The proper course to pur-

sue before introducing a boiler compound is to have an analysis of the water made, and only after the mineral ingredients and the quantity of each is known can the correct method of treatment in order to avoid boiler trouble be applied.

All reputable manufacturers of boiler compounds nowadays make it a practice to analyze the water (whenever the composition is unknown) before furnishing the compound, and as a result, the boiler compound is prepared specially to suit the requirements of the individual water.

The boiler compound which is to be used in the brewery, distillery, vinegar and yeast factory, ice-plants and by manufacturers of food products, where live steam is used for heating or cooking purposes, must also be guaranteed harmless as far as the *effect upon the taste or odor* of the steam is concerned. This is imperative, since otherwise the quality of the product which is manufactured can easily be impaired.

It would be futile to give the compositions of even the most important boiler compounds on the market, since they vary considerably, according to the type of water for which they were compounded. Below, however, are given the most important chemicals or substances, which are the main active constituents of practically all of the boiler compounds in common use at present, whose action is a *chemical one,* viz., neutralize any corroding ingredients of the water and change the incrustating salts that form a hard scale into new salts which either produce a comparatively soft scale or cause mud.

Sodium Carbonate (Soda Ash).—This chemical is used very extensively in the manufacture of boiler compounds, in fact it is the principal scale-preventing constituent of the greater number of them. It is effective for changing calcium sulphate (gypsum) to calcium carbonate and in that way prevents the formation of a hard scale.

Sodium carbonate also neutralizes any acids in water and reacts with magnesium chloride, which has a corrosive effect on iron, changing it to magnesium carbonate. Calcium chloride and magnesium sulphate, although readily soluble in water, are also precipitated as carbonates by sodium carbonate.

It has no chemical action upon calcium and magnesium bi-

carbonates, but practical experience in boiler operation has shown that when the water in the steam boiler reacts slightly alkaline from sodium carbonate, the precipitated calcium and magnesium carbonates have more of a tendency to remain in the boiler water as mud or sludge than towards the formation of a soft scale.

Sodium carbonate, or rather as commercially termed, soda ash, is a very effective scale and corrosion preventative when the boiler feed-water contains calcium sulphate, magnesium or calcium chloride, or reacts acid. The amount needed must be carefully determined by the chemist and the addition of excessive quantities must be avoided, because under such conditions foaming or priming of the water in the boiler can readily take place. Packing and gaskets, also, may become affected, resulting in leaks, and the gauge glasses frequently become milky due to the action of the alkali on the glass.

Sodium Hydrate (Caustic Soda).—This chemical is a much stronger alkali than sodium carbonate, and is one of the ingredients of many well known boiler compounds. When added to a boiler feed-water, it first combines with carbonic acid gas, usually the excess molecule of the calcium and magnesium bi-carbonates. These are then precipitated as carbonates and the caustic soda is changed to sodium carbonate. The latter acts upon the calcium and magnesium sulphates and chlorides, as outlined under sodium carbonate. If sufficient carbonic acid is present, the calcium and magnesium sulphates and chlorides are thrown out of solution as hydrates of these elements. Caustic soda in excessive quantities gives rise to foaming even more readily than sodium carbonate, especially in the presence of a considerable amount of organic substances or saponifiable fats or oils.

Calcium Oxide (Burnt Lime) is rarely present in boiler compounds. This chemical when slaked in water is frequently added together with sodium carbonate to the feed-water when softening it in external softening apparatuses, and should only be used in this way. It combines with the extra molecule of carbonic acid gas of the calcium and magnesium bi-carbonates, precipitates them as carbonates, and is itself thrown out of solution as calcium carbonate.

Barium Hydrate (Caustic Baryta) is sometimes used in external water-softening devices. It is also employed in some

types of mechanical boiler cleaners. The barium hydrate takes up carbonic acid gas from the bi-carbonates, precipitates them and at the same time falls out of solution as barium carbonate. When added directly to the water in the boiler, in the absence of carbonic acid gas, the barium combines with the sulphates, forming barium sulphate. In that way not only the calcium sulphate, but also sodium or magnesium sulphate, is eliminated. An important characteristic of the insoluble barium salts, viz., the carbonate and sulphate, is the high specific gravity, and on account of this they settle very rapidly.

Barium Chloride is sometimes used in place of the hydrate, but particularly when the feed-water is alkaline due to the presence of sodium carbonate. Its action is to eliminate the latter, forming insoluble barium carbonate. Barium salts are employed comparatively little nowadays, mainly on account of their cost.

Sodium Fluoride is seldom used, mainly on account of the high price. It precipitates the soluble calcium and magnesium salts as fluorides, and these are of such a light, flocculent, consistency that they can easily be removed through the blow-off valve.

Tri-Sodium Phosphate.—This chemical represents a very strongly alkaline sodium phosphate and has been used for many years with very good results. It precipitates some of the soluble calcium and magnesium salts as phosphates, which are very flocculent and easily settle to the bottom of the boiler, from where the deposit can be removed by expelling small quantities of water through the blow-off valve at regular intervals. Besides this action of the phosphate, the additional combined sodium oxide exercises a softening action in the same way as caustic soda does, but on account of the presence of the flocculent calcium and magnesium phosphates, the precipitated carbonates, as well as any other scale-producing solids, are carried down together with the phosphates more easily than when either soda ash or caustic soda is employed alone.

Tannic Acid and Tannates.—Tannic acid either in the pure state or in the form of bark extract is seldom used alone nowadays, and its employment is not to be advised because the

acid will attack iron. However, when present as sodium tannate, especially together with an excess of caustic soda or sodium carbonate (soda ash), very good results as to scale prevention can be obtained; in fact several of the most popular boiler compounds are composed chiefly of strongly alkaline sodium tannate, made by mixing bark extract (generally from oak or hemlock bark) and caustic soda or soda ash.

Sugar is sometimes used in the preparation of some boiler compounds. As a rule, a low grade molasses or the refuse liquor obtained in the sugar refinery is employed. The sugar combines with the calcium and magnesium salts forming saccharates, which are quite flocculent in nature.

Sodium Silicates or **Water Glass,** being distinctly alkaline in reaction, give very good results for preventing scale formation. They combine with the soluble calcium and magnesium salts of the water in the boiler and form calcium and magnesium silicates. The latter are very fluffy or rather gelatinous, and can easily be expelled when forcing out small amounts of water through the blow-off valve at frequent intervals.

Permutite is the name of a new material recently used as a scale preventative, and the process of its employment, patented in Germany, has been introduced in many large steam plants in that country, where it is giving good results. The material is an aluminium-sodium silicate obtained by smelting together clay, silica, soda and several minor substances. The resulting product is an artificial zeolithe and is placed on the market in a finely crushed condition, the grains usually being about the size of sea-sand. The sodium silicates really are the active ingredients, and when the feed-water slowly flows through a layer or bed of the Permutite, the calcium and magnesium salts react with them and are eliminated from the water. After using for a certain quantity of water, the Permutite must be regenerated, which is accomplished by passing a sodium chloride solution through it. By so doing the accumulated calcium and magnesium salts are removed.

Zinc is recommended especially for preventing corrosion when the boiler water contains a considerable amount of chlorides and particularly magnesium chloride. A scale-preventing action

also is claimed for it. Zinc slabs $12 \times 6 \times \frac{1}{2}$ inches, made of rolled zinc, usually are suspended by means of iron straps, from the stays or tubes. Practical experience has shown numerous instances where zinc was not effective in preventing corrosion or scale formation but on the contrary gave the opposite results.

Besides the before-mentioned substances which have a chemical action upon the scale-forming ingredients of the boiler feedwater, there are other substances employed as scale-preventatives which have purely a mechanical effect in this direction. Such intermix with the incrustating solids when these come out of solution in the boiler, and in that way prevent them from caking together and baking on the boiler sheets and flues; or they will cover the iron with a very thin coating, so that the scale can not adhere to the iron surfaces. Among such substances may be mentioned the following:

Starch.—The addition of potatoes to the water in the boiler is one of the oldest methods of scale prevention. The resulting pastified starch envelopes or surrounds the small crystals of precipitated calcium and magnesium salts, and prevents them from caking together. The thus accumulated mud or sludge was removed to a great extent by blowing off small quantities of water. If too much starch was introduced, trouble usually resulted on account of foaming or priming.

Starch in the form of corn meal or ground corn is often a constituent of boiler compounds, but usually the main scale-preventing ingredients of such a compound are soda ash or caustic soda and sodium tannate, originating from bark extract. In such cases the addition of the starchy material is made mainly in order to cause the compound to be of a more gelatinous or jelly-like consistency.

Wood Fibres are usually added in the form of fine sawdust or powdered bark. The particles of cellulose are supposed to furnish the nucleus or center to which the precipitated solids adhere and in this way retard the incrustation. Besides this mechanical action, the sawdust or bark furnishes small amounts of tannates which also have a beneficial, although insignificant, effect. In former years, many boiler compounds were nothing else but a

mixture of a low grade soda ash and fine sawdust usually from oak wood or powdered oak or hemlock bark.

Kerosene; Petroleum Oils.—Kerosene and sometimes other light petroleum oils have been used successfully as scale preventatives. They cover the precipitated solids with a minute oily coating, thereby checking them from aggregating in the form of scale. The use of kerosene or other light petroleum oils is not to be recommended when live steam is employed for direct heating, like in the brewery, distillery and in the manufacture of distilled water ice or of food products.

Under no circumstances should saponifiable oils or the heavier petroleum oils be employed. As already stated earlier in this chapter, such oils mix with the scale and make it practically waterproof. When this condition exists, a boiler explosion can readily occur.

Graphite.—So-called boiler graphite has been placed on the market very recently and is giving very good results wherever given a trial. It is prepared from a high grade natural graphite containing about 85% of graphite carbon. The mined graphite is ground to the form of an impalpable powder having a sieve test of practically 100% through a 240 mesh sieve. The graphite consisting of such very finely divided particles has the peculiar property that it is greasy to the touch, and has an entirely mechanical effect as far as scale prevention is concerned. It intermixes with the crystals of incrustating salts and causes them to remain in the boiler water in the form of a sludge, which can readily be removed by blowing off. The graphite furthermore causes the iron surfaces inside of the boiler to become covered with a fine coating thereof, and has such a lubricating action that the scale will not adhere to and bake on them under the usual boiler conditions. Tests made with badly scaled boilers have shown that the finely divided particles of graphite gradually work their way underneath the scale and finally loosen it from the iron.

Boiler graphite should be fed into the feed-water preferably after it has passed through the feed-water heater, so that all of it actually enters the boiler. Only small quantities are needed for effective results, the claim being made that one ounce per 1,000 gallons of water evaporated is sufficient for even medium hard

boiler waters (15-20 grains incrustating salts per gallon). If the feed-water contains magnesium chloride, free acids, or any ingredients having a corrosive effect on boiler iron, it is recommendable to add soda ash or caustic soda in sufficient quantities to render the water in the boiler slightly alkaline at all times.

Application of the Boiler Compound.

Numerous methods of adding the boiler compound are practiced. These may be divided into two classes, viz., *continuous* feeding and *intermittent* feeding. According to the first one, small quantities of the compound are fed at regular short intervals. The intermittent method of introducing a boiler compound is most commonly employed, and according to this, a certain amount is added once a day in the case of small boilers and twice or oftener in large boilers. The quantity introduced depends on the amount and the nature of the ingredients of the water. Since nearly all boiler compounds are alkaline in reaction, a good practice is to test the water in the boiler with *red litmus* paper. This can be done by taking a sample of the water through a gauge cock, collecting it in a clean glass and then immersing the red litmus paper. As long as the paper turns distinctly blue, it is an indication that sufficient compound is present to neutralize any corroding substances and to change those producing a hard scale to such as are less dangerous, which is the main object of adding the boiler compound. The resulting sludge or mud can then be eliminated to a great extent by expelling 5 to 10 gallons of water every 4 to 6 hours, through the blow-off valve of the boiler. In this way, the amount of incrustating substances in the boiler will be kept at a minimum, and they, therefore, cannot bake on to the boiler iron.

It is self-evident that the best method of adding the boiler compound is to introduce it continuously by means of an independent small pump or an automatic feeding device, into the feed-water before or at the time the latter enters the heater, since by so doing there is a combined and more effective action of both the chemical and the heat. If the water is then filtered before it enters the boiler, either by means of a filter such as is a part of feed-water heaters of the open type, or through a separate filter, practically all of the insoluble particles are removed before the feed-water enters the boiler.

Water Softening Apparatus.

Where large quantities of water, especially one rich in incrustating solids, are used daily for boiler feeding purposes, the common practice is to eliminate the scale-producing substances by passing the water through a water softening apparatus. Such treatment generally also neutralizes any corroding substances which may be present in the water.

Such water softening is accomplished by the addition of certain chemicals, usually soda ash or caustic soda, and often together with slaked lime or caustic baryta. In some instances, this treatment is applied to the cold water and the precipitated solids are removed by either sedimentation or filtration, whereas in others, the chemicals are added to the heated water, or the water is heated after adding the chemicals, and the insoluble substances are allowed to settle or are removed by means of a filter. The hot treatment, under ordinary circumstances, is more effective than the cold, but if the latter is employed, equally good results will be obtained, if the water is heated to a temperature of about 210° F. in a feed-water heater, especially one of the open type, before it enters the boiler.

The apparatus employed for external softening of the boiler feed-water vary greatly in construction. They are to be classified as *intermittent* and *continuous* as to their operation.

The simplest form of an *intermittent* type water softening apparatus consists of two or more large tanks holding a sufficient quantity of water for a day's run. The softening chemicals are added to the water while filling the tank, and as soon as it is filled, steam or air should be forced into the water for 10 to 15 minutes, in order to mix thoroughly the water and the chemicals. It is then allowed to stand for 6 to 12 hours, and during this time the precipitated solids settle to the bottom. The clear water above the sediment is now drawn off, preferably taking it from the top, which can be accomplished by having a hose attached to the outlet pipe leading to the boiler feed-water heater or pump, that extends into the tank. This hose should be coupled to a float device which holds it submerged about 4 inches below the surface of the water. Since the top layers of water become clear most rapidly, such clear water will be drawn off first by means of this arrangement.

The chemicals generally employed for this treatment are soda

ash and slaked lime. The amounts that must be added depend upon the composition of the water, and an accurate analysis must therefore be available. The chemist making the analysis will state the amounts of the softening chemicals that are to be added per 1,000 gallons of the water, if requested to do so.

Water softening apparatus of the *continuous* type are most commonly installed nowadays, in preference to those of the intermittent type, principally on account of the more convenient opera-

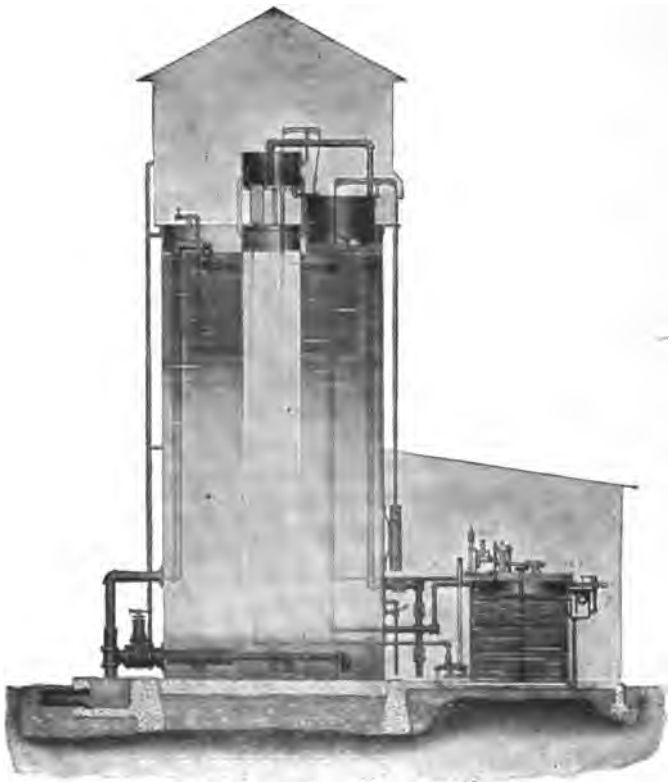


Figure 90. Graver Water-Softening Tank.

tion, and because they require only ⅓ to ½ as much ground space. There are a number of high grade water purifiers of this type in use, but space will not allow giving a detailed description of the construction and manipulation of all. In such devices, the water is softened according to either the cold or hot process.

The *hot process* water softening devices really are a combination of the cold process apparatus and a feed-water heater. Usual-

ly only soda ash or caustic soda are added to the raw water, in order to precipitate the calcium and magnesium chlorides and sulphates. The bi-carbonates of calcium and magnesium are made insoluble by the heat. In the cold process devices, practically all of these substances are eliminated by the chemicals, and the water has been softened to such an extent that the heating in the feed-water heater does not remove any further appreciable quantities.

The *cold process* apparatus generally consists of a high cylindrical tank, inside of which is a cylindrical pipe, extending almost to the bottom. This usually is called the reaction tank. On the top of the tank is mounted a mechanical device that automatically discharges a certain quantity (which can be regulated) of the raw water and the softening chemicals, usually soda ash and slaked lime. The mixing frequently is facilitated by a stirring device. The treated water then flows downwards through the inner cylinder pipe, viz., the reaction tank, and passes upwards in the outer space, i. e., between the walls of the small and the outside cylinder. On account of the slow flow of the water, the greater portion of the precipitated solids settles at the bottom of the tank. Those particles that do not settle are removed by the filter, containing quartz, gravel, excelsior or felt, which is placed several feet from the top of the cylinder. The water is then drawn from above the filter, passed through the feed-water heater, from where it enters the boiler. A typical cold process water softener that is giving very good results in practice is shown by Figure 90.

The various types of cold process water softeners differ prinpally in the construction of the mechanical device which measures off the amount of the softening chemicals to be introduced per certain quantity of the raw water. Many ingenious arrangements have been patented and are giving very satisfactory results. The main precautions that must be observed in order to soften the water successfully are:

1) That the device measures accurately the proper amount of chemical needed for the quantity of water flowing into the softener at a certain speed.

2) That the water and the chemicals are intimately mixed.

3) That sufficient length of time is allowed so as to obtain a complete reaction. In cold weather, two or four hours usually

are needed. Heat greatly facilitates the chemical change, and the hotter the water the faster will the incrustating solids be eliminated.

4) That ample sedimentation and clear filtration is had, so that a clear water leaves the softener.

Removal of Boiler Scale.

Steam boilers are shut down from time to time in order that they may be cleaned. When so doing, the fires should be drawn and the boilers allowed to cool slowly in the air. Often it is only necessary to wash them out by means of a powerful stream of water, since the insoluble mineral substances originating from the feed-water are present in the form of a sludge or soft mud. If scale has accumulated, it must be removed before putting the boiler into operation again. If the scale is in the lower part of the boiler, where access is very difficult, a boy frequently is sent into the boiler and the scale removed by scraping it off with suitable implements. Where the scale has deposited on or in the flues or tubes, it can be dislodged by using a so-called *boiler-tube cleaner*. These devices are operated by steam or compressed air, and in the head of one type of this tool there usually is a hammer or plunger, which strikes blows in very rapid succession. Such hammering or the vibration produced by it in the tube causes the scale to fall off. Another type of tube cleaner has a cutting device in the head of the tool, which cuts or scrapes off the scale.

A very effective *chemical method* for the removal of a very hard boiler scale is to fill the boiler with water and add sufficient caustic soda so that a 1½ to 2% solution of the latter results. A small fire is then built and the boiler allowed to steam, under atmospheric pressure, for from 24 to 72 hours, depending on the thickness of the accumulated scale. This means that the alkali solution should boil slowly. The caustic soda will cause the scale to disintegrate, and in time it will loosen completely from the boiler iron. The solution is then run out and the boiler thoroughly washed. When employing this method the precaution must be observed to examine carefully, and if necessary replace any defective packing or gaskets, because they may have been affected by the alkali.

Steam Engines

Although during the last decade the steam turbine, the gas engine, the oil engine, and electrical power have in many instances replaced the reciprocating steam engine, the chances are that the reciprocating engine will maintain its position as the most common prime-mover for some time to come.

There are various methods of classifying steam engines, such as *simple* and *compound, condensing* and *non-condensing, reciprocating* and *rotary, plain slide-valve* and *automatic cut-off,* etc. All of these classes are explained later in their proper places.

The Plain Slide-Valve Engine.

The slide-valve engine is very simple in construction, but not highly economical. All the parts of this type of engine, described below, will also be found in any other type of double-acting, reciprocating steam engine, either in a modified form, or with additional parts for speed control, increased steam economy, etc.

Figure 91 shows a plan section of a plain slide-valve engine. In the *Cylinder 1,* the *Piston 2* moves forward and backward, and thereby produces work. The piston must fit steam-tight and still move easily. In order that this may be accomplished, the wearing surface is made up of steel spring rings.

The *Steam-Chest 3* receives the steam from the steam pipe, and is always under full steam pressure. By moving forward and backward, the *Slide-Valve 4* brings each of the *Steam-Ports 5* alternately in connection with the steam chest and with the *Exhaust-Outlet 6.*

In the position shown in Figure 92, the left steam port and the left side of the piston are under steam pressure, while the right steam-port and the right side of the piston are under atmospheric pressure (in practice, back pressure must be con-

sidered). It is evident that these conditions will cause the piston to move from the left towards the right.

The *Cylinder-Heads 7 and 8* close the cylinder at the ends. The *Piston-Rod 9* transfers the motion of the piston to the *Cross-Head 10.* In the cylinder-head 8, there is an opening for the piston, which is made steam-tight by means of the *Stuffing Box and Gland 11.* The piston-rod is made of steel, and is rigidly connected to the piston and the cross-head. The cross-head is made of strong cast iron, and by means of the *Guides 12,* is made to move in a straight line, without any deviations.

By means of the *Connecting Rod 13,* the reciprocating motion of the cross-head is transmitted as rotary motion to the *Crank 14* and *Shaft 15.* The connecting rod swings on the *Wrist Pin 16* in the cross-head and acts on the crank through the *Crank Pin 17.* The connecting rod is a steel forging with adjustable bearings. The crank pin is fixed rigidly on the crank, and this is keyed to the shaft. It is evident that the travel of the piston during one stroke is equal to twice the length of the crank.

The *Fly-Wheel 18* is keyed to the shaft. The necessity of having a fly-wheel is shown later (see page 316).

The *Eccentric 19* is also keyed to the shaft, and through the *Eccentric Rod 20* transmits its rotary motion as reciprocating motion to the *Valve-Stem 21,* and to the *Slide-Valve.* A guide for the valve-stem is shown at *22* and a small cross-head at *23.* In one end of the steam chest there is an opening for the valve-stem, which must be made steam-tight by means of a *Stuffing Box and Gland 24.*

Finally, there is the *Frame,* made of cast iron, which holds the different parts together and carries them at a convenient height above the floor. A complete description of all these parts does not come within the scope of this book. However, the principles involved are given in the following pages.

Action of Plain Slide-Valve.

The following description, used in connection with the illustrations, shows the action of the slide-valve.

In Figure 92 the piston is shown just at the start of its forward stroke. The movement of the slide-valve has been such that steam has shortly before begun to be admitted to the

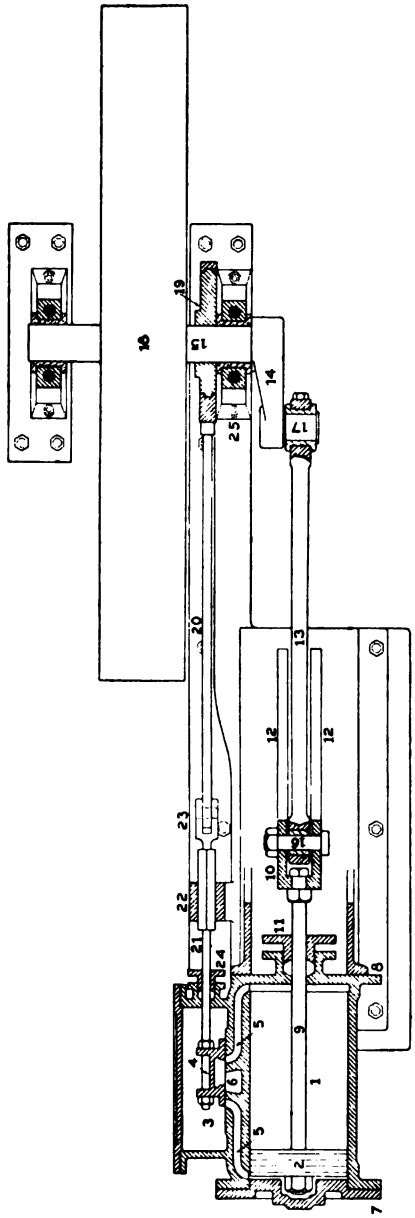


Figure 91. Slide Valve Engine.

space between the piston and the *head end* of the cylinder. Thus the *point of admission* occurs before the beginning of the stroke, there being a small opening to steam at this time, as explained under Lead, page 311. The *crank end* of the cylinder has opened to the exhaust.

In Figure 93, the steam port and the exhaust port are both fully open, the piston is still in the first part of its travel, and the valve is almost in its extreme position towards the crank end.

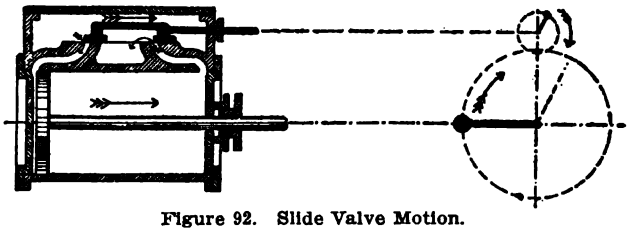


Figure 92. Slide Valve Motion.

In Figure 94, the steam port has been closed, this being called *the point of cut-off,* and only the expansive force of the steam pushes the piston ahead. The space in front of the piston is still open to the exhaust. The valve is now moving in the opposite direction to that of the piston.

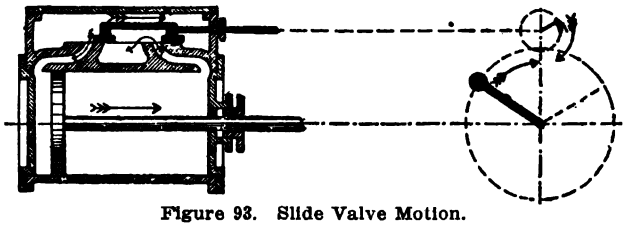


Figure 93. Slide Valve Motion.

In Figure 96, the valve is in its middle position, and the piston is nearing the end of the stroke. The exhaust-port has been closed, the point where this occurred being called the *point of compression,* the exhaust steam in front of the piston being subjected to compression. In Figure 95, the valve has passed its middle position and is just opening connection between the exhaust-outlet and left side of the piston, which position is called the *point of release.* The piston is near the end of its stroke, and a very slight further advance will bring it to the point of ad-

mission for the return stroke; another small advance will bring it to the extreme position at the crank end of the cylinder.

Referring to Figure 96, AB is called the *outside lap,* and CD the *inside lap;* the outside lap being the amount the edge of the valve overlaps the outside edge of the steam-port, and the inside lap being the amount the edge of the valve overlaps the inside edge of the steam-port. If there were no lap, steam

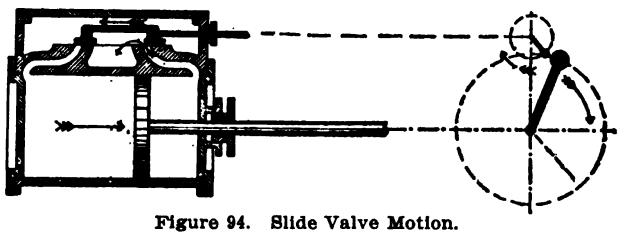


Figure 94. Slide Valve Motion.

would be admitted during the whole stroke, there being no cut-off, with consequent waste of steam.

In Figure 97, EF is called the *lead,* and it indicates the amount of opening for the live steam at the start of the stroke. If there were no lead, the steam-port and exhaust-port would not open until the piston starts on its stroke. Thus, live steam would not immediately fill up the clearance space, and the piston would make the first part of its stroke without full steam

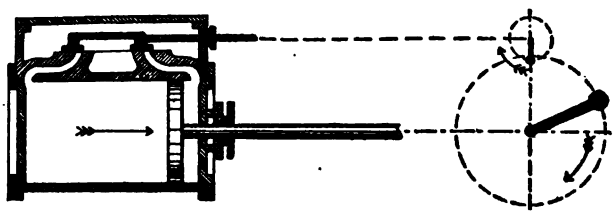


Figure 95. Slide Valve Motion.

pressure behind it. Compression would of course assist towards raising the pressure at the start, and could be carried so far as to create the full live steam pressure in the clearance space. However, experience has proven that a liberal lead, giving a clearance space filled with steam at full pressure at the beginning of the stroke, is essential for good running, and the more so the higher the speed of the engine.

The lead in stationary engines is on an average about 1/16 of an inch for medium speed, and greater for higher speeds. By studying the diagrams, it becomes evident that, in order to give lead, and to counterbalance the lap, the eccentric operating the slide-valve must be more than 90 degrees ahead of the crank, as the shaft turns (see Figures 92 and 95). The number of degrees it is ahead of 90 degrees is called the *angular advance*.

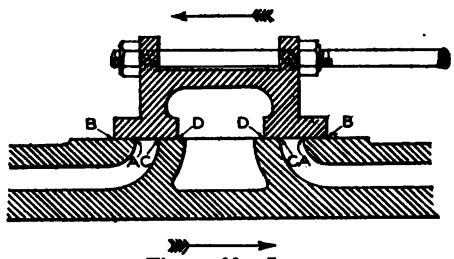


Figure 96. Lap.

If the angular advance is kept the same, and the outside lap increased, an earlier cut-off and more complete expansion is caused, while a diminished outside lap means later cut-off and less expansion. Increasing the inside lap produces greater compression and later release, while decreasing the inside lap makes

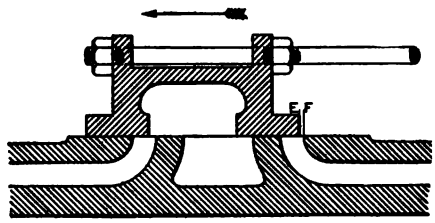


Figure 97. Lead.

compression smaller and causes earlier release. In other words, the outside lap controls cut-off and expansion, the inside lap release and compression. For these reasons, the outside lap is often called the *steam lap,* while the inside lap is called the *exhaust lap*. The outside lap has no influence on compression, and the inside lap none on expansion.

The Eccentric.

The eccentric, which gives motion to the slide-valve, consists of a circular disc fastened to the shaft, as shown in Figure 98.

The radius of the eccentric, that is the distance from the center of the shaft to the center of the eccentric, is commonly called the *eccentricity.*

The *throw* of the eccentric is given by some authors as equal to the radius of the eccentric, and by others as equal to twice the radius.

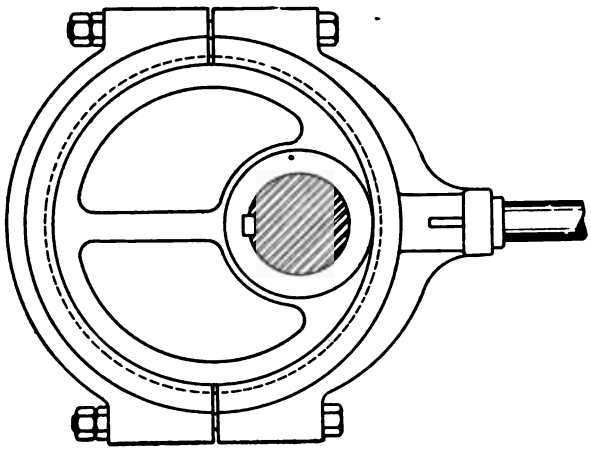


Figure 98. Eccentric.

The *travel* of the valve is the distance the valve moves back and forth during one stroke of the piston. Consequently the travel is either equal to the throw, or to twice the throw of the eccentric, according to which meaning is taken for the throw. The travel of the valve must be equal to two times the port opening plus two times the outside lap.

Double-Ported Valves.

In large cylinders, where the travel of the valve would have to be large, the double-ported valve is used, by which means the travel of the valve is reduced by one-half.

Dimensions of Steam-Ports.

The area of a steam-port should be greater than that of the steam-pipe. Experiments have shown that a steam-pipe cross-section of from 6/10 to 9-10 of the steam-port area, according to prevailing conditions, gives free admission to the steam.

The length of the ports is usually about 8/10 of the diameter of the cylinder.

Clearance.

When the piston is in its extreme position at either end, there is still a small space left between the piston and cylinder head (see Figure 92). This space is necessary, so as to allow changing the length of the connecting rod or piston-rod without causing the piston to come in contact with the head, and also so as to avoid any danger due to water left in cylinder, which is thus given space in which to collect. This distance is called the *piston clearance*. It may be easily measured after placing the engine on dead-center.

The volume of this space, plus the volume of the steam-ports at the same end is called the *clearance volume* or *clearance*. It may be expressed in various ways, but is generally given as a per cent of the *piston displacement*, that is a per cent of the volume swept through by the piston during one stroke. It varies from 1 to 10 per cent.

As mentioned before, when the exhaust port is closed (see Figure 95), *compression* of the remaining steam in front of the piston begins. At the end of the stroke this steam is compressed into the clearance space. Its pressure is thereby considerably increased (the more so the earlier compression begins), and it forms a *cushion* between piston and cylinder-head, preventing to some extent jerks and knocks.

The clearance volume may be measured by placing the engine on dead-center, and filling the space between the piston and the head of the cylinder, and also the port leading to this end, with water. From the weight of the water used, the volume can be calculated. The piston displacement can readily be calculated from the diameter of the cylinder and the length of the stroke.

Example.—In a 12×18 inch engine, 4 pounds of water were required to fill the clearance space and steam-port. What is the per cent of clearance?

Solution.—Cylinder volume $= 12 \times 12 \times 0.7854 \times 18 = 2{,}035.8$ cubic inches.

Since 1 gallon $= 231$ cubic inches, this equals $\frac{2{,}035.8}{231}$ $= 8.81$ gallons.

Since one gallon $= 8\ 1/3$ pounds of water of average temperature, 4 pounds $= \frac{4}{8.33} = 0.48$ gallon.

Hence, per cent clearance $= \frac{0.48 \times 100}{8.81} = 5.4\%$.

Setting Slide Valves.

The only parts that can be adjusted, when setting the valve of a plain slide-valve engine, are the valve-rod and the angular advance of the eccentric.

The valve-rod should be so adjusted as to make the valve travel an equal distance each side of the mid-position. This must be done before the eccentric is set. It is done by loosening the eccentric on the shaft and turning it around until it gives the maximum port opening, first at one end, and then at the other. If these port openings are not equal, they must be made so by changing the length of the valve-rod by half the difference between the port openings. When the above adjustment has been made, set the engine on a dead center, as described below, and give the valve the proper lead by turning the eccentric ahead of the crank in the direction the engine is supposed to run.

Setting Engine on Dead-Center.

The proper method for setting the engine on dead-center is as follows: Turn the engine in the direction in which it is supposed to run, until the cross-head is about one-half inch from the end of the stroke. Make a mark on the cross-head corresponding to a mark on the guide, and make a mark on the fly-wheel rim corresponding to a fixed point near the floor, which is generally designated by a nail driven into a 6 by 8-inch block of wood. Then turn the engine ahead until the

mark on the cross-head corresponds to the same mark on the guide, and make another mark on the fly-wheel rim, corresponding to the same fixed point. We then have two marks on the fly-wheel, and if we make another mark half-way between these two marks, and turn the fly-wheel back until this center mark meets the fixed point, then our engine is on the dead-center. A mark should now be made on the guide to correspond with the mark on the cross-head. The dead-center of the other end should be found in the same manner and likewise indicated on the guide.

Fly-Wheels.

The object of a fly-wheel is to help the engine over the dead centers, and to give it a steady motion.

Referring to Figures 92 to 95, it will be realized, that when the crank is in the position shown in Figures 93 and 94, the pressure behind the piston will produce turning of the crank and shaft. In the position shown in Figure 92, however, no amount of pressure behind the piston will cause the crank to turn, this position being called the dead-center. The same is true at the other dead-center. However, when a heavy fly-wheel is keyed on to the shaft, it gathers so much momentum, while the piston and crank pass through the positions shown in Figures 93 and 94, that it keeps on moving through the dead-center, turning the crank and pushing the piston.

However, the valuable effect of the fly-wheel is not solely confined to helping the moving parts of the engine over the dead-centers. Speaking more in general, the force with which the piston turns the crank and shaft varies all the time. At the start of the stroke it is greatest, and of course decreases as expansion goes on in the cylinder. As shown above, the force with which the shaft is turned, depends not only on the force pushing the piston ahead, but also on the momentary position of the crank. At one point only, will the momentary work given off be equal to the work absorbed, while before this point is reached, more work will be given off than is absorbed by the machinery, and beyond this point, less. Therefore, the superfluous work given off by the engine will be used to increase the speed of the fly-wheel, while at another part of the stroke, the fly-wheel, by slowing down, gives off energy enough to supply

the amount of work which the engine is short. The fly-wheel thus becomes an energy reservoir. It is evident, that some alternating speeding up and slowing down of fly-wheel and shaft, during each stroke, is inevitable. However, by increasing the weight or the diameter, of the fly-wheel, or both, these speed variations may be made so small as to be of no practical significance. How far to go in this respect depends in every case, on the speed allowed and on the nature of the machinery that is to be driven. The higher the speed, the smaller the fly-wheel can be made for equally good speed regulation. The requirements for uniform, steady motion of the shaft of course vary considerably in different plants.

Under actual conditions, the load on the engine may be practically uniform for hours, and then in a few minutes decrease say, 30 to 40 per cent and stay there. In this case, the fly-wheel is also useful, as it prevents the engine from speeding up immediately. However, if stroke after stroke the engine would give off so much more work than the driven machinery absorbs, the speed of the fly-wheel and of the shaft would evidently keep on increasing, slowly and steadily, and finally, if not absorbed, become dangerous to the plant. Here is where the *governor* offers an automatic remedy (see page 320).

Besides serving its main purpose, the fly-wheel may in some case conveniently be used as a belt pulley; in other cases the shaft is made longer and provided with a special belt pulley, or it may be coupled directly to the shaft of the machine to be driven.

The effectiveness of a fly-wheel in steadying the motion of an engine is proportional to the weight of its rim, to the square of its diameter, and to the square of the number of revolutions per minute. Increasing the diameter therefore, saves weight. In the construction of a fly-wheel, the material should be concentrated as much as possible in the rim.

Calculation of Weight of Fly-Wheel.—There are various formulas for calculating the weight of fly-wheels. Thurston gives an approximate formula for ordinary non-condensing engines with cut-off of $^1/_5$ to $^1/_3$, as follows:

$$W = \frac{c \times A \times S}{R^2 \times D^2}$$

W = weight of rim of wheel in pounds.
A = area of piston in square inches.
S = length of stroke in feet.
R = number of revolutions per minute.
D = outside diameter of wheel in feet.
c = 10,000,000 to 15,000,000, on the average 12,000,000.

Example.—In a simple non-condensing engine, the diameter of the piston is 12 inches and the length of stroke 24 inches (2 feet). What should be the weight of the fly-wheel if it is 6 feet in diameter, and makes 140 revolutions per minute?

Solution.—$$W = \frac{12 \times 12 \times 0.7854 \times 2 \times 12{,}000{,}000}{140 \times 140 \times 6 \times 6} = 3{,}847.$$

The weight of the wheel should be 3,847 pounds.

Speed of Fly-Wheel.—When a body rotates around an axis, the centrifugal force becomes greater the higher the rotatory speed, and may become so great that the body is torn apart.

The portion torn loose by the action of this centrifugal force would follow the direction of the tangent at the point where the rupture occurred. The resistance which the material of a fly-wheel offers to the centrifugal force depends upon its so-called tensile strength, which varies with the nature of the material. Taking the tensile strength of cast iron, for instance, at 10,000 pounds per square inch, this means that a force of 10,000 pounds per square inch of cross-section is necessary to tear the cast iron apart.

For every material there is a certain speed at which breakage will occur, regardless of the quantity of material used. For fly-wheels with solid rim this speed can be found according to the following formula:

$$V = 1.6 \times \sqrt{\frac{T}{W}}$$

V = linear velocity of the rim of the wheel in feet per second, at which breakage occurs.
T = tensile strength of one square inch of material.
W = weight in pounds of one cubic inch of material.

If in this formula the safe strength is used instead of the tensile strength, the result will be rim speed in feet per second at which the wheel can be run with safety. The factor of safety generally used is 10%.

Example.—At what speed can a cast iron fly-wheel be run with safety, if the weight of one cubic inch of cast iron is 0.26 pound, its tensile strength 10,000 pounds, and if we assume a factor of safety of 10%?

Solution.—$V = 1.6 \times \sqrt{\frac{10{,}000 \times 0.10}{0.26}} = 100.$

The safe speed is 100 feet per second.

If instead of cast iron, *cast steel* having a tensile strength of 60,000 pounds and weighing 0.28 pound per cubic inch, is used in the construction of the fly-wheel, the safe rim speed will be:

$$V = 1.6 \times \sqrt{\frac{60{,}000 \times 0.10}{0.28}} = 234 \text{ feet per second.}$$

For rolling-mill engines, fly-wheels have been constructed of *wrought iron* I-beams. They are considerably stronger than cast iron fly-wheels, since the tensile strength of wrought iron is as high as 40,000 pounds per square inch.

As stated, the above formula applies only to wheels with solid rims. If the wheel is made in halves or sections, it is considerably weaker, and the efficiency of the rim joints must be taken into consideration. For belt wheels with flanged and bolted rim joints located between the arms, the joints average only about $\frac{1}{5}$ the strength of the rim and never more than $\frac{1}{4}$.

If the rim is thick enough to allow the joint to be reinforced by *steel links* shrunk on, as in heavy balance wheels, $\frac{1}{3}$ the strength of the rim may be secured in the joint. This construction, however, cannot be applied to belt wheels with thin rims

Horse-Power Required to Drive Fly-Wheels.—The horse-power required to drive fly-wheels is expressed by the following formula:

$$\text{H. P.} = \frac{f \times w \times R \times d \times 0.26}{33{,}000}$$

H. P. = horse-power required.
f = coefficient of friction = 0.03 to 0.25, average 0.08.
w = weight of wheel in pounds.
R = revolutions per minute.
d = diameter of shaft in journal in inches.
0.26 = weight in pounds of 1 cubic inch of cast iron.

Example.—A shaft 10 inches in diameter, having a fly-wheel weighing 12,000 pounds, makes 75 revolutions per minute. As-

suming a coefficient of friction (f) of 0.08, which is about the average, how many horse-power are required to drive the fly-wheel?

$$\textit{Solution.}\text{—H P.} = \frac{0.08 \times 12{,}000 \times 75 \times 10 \times 0.26}{33{,}000} = 5.67.$$

There are 5.67 horse-power required to drive the fly-wheel.

Engineers often have the misconception that power for driving fly-wheels is required only in starting, but the above formula shows that power is required continuously.

Governors.

While the fly-wheel balances the speed variations, which would otherwise occur regularly during each stroke, the governor

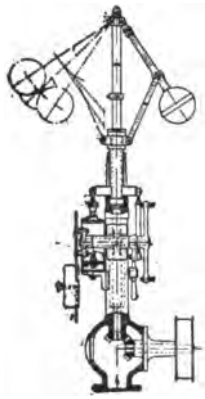


Figure 99. Throttling Governor.

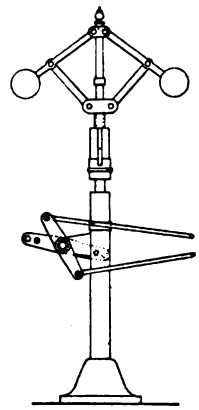


Figure 100. Corliss Ball Governor.

takes care of variations in speed caused by changes in the load or in the boiler pressure. Thus, in order to make the engine run with a fairly uniform speed, an almost continuous process of adjusting the work given off during a certain number of strokes to the work consumed is required. Whatever the construction of the governor is, its action is always either to make the cut-off of the steam occur earlier or later in the stroke, or to change the pressure of the steam before entering the steam-chest.

In this respect, steam engines are therefore classified as automatic cut-off engines and as throttling engines, the latter being nowadays unimportant except in small sizes.

The principle of centrifugal force as embodied in the old fly-ball governor of Watt has been more resorted to than any other. Many improvements on governors have been made since Watt's time, but the principle remains the same, as does also the main defect of the original device which so materially interferes with its efficiency, namely, friction.

The different designs of modern governors are manifold, and their description goes beyond the scope of this book. Only some of the most important ones will be mentioned here.

Figure 99 shows a *throttling governor* in which the vertically moving rod raises or lowers a steam valve, and thus admits more or less steam, according to the load on the engine.

In Figure 100 is shown a governor directly attached to the valve motion such as is used on Corliss and Poppet valve engines.

The *fly-wheel or shaft governor* used on high speed engines is illustrated and described under High-Speed Engines, page 353.

Inertia Governor.—This type of governor designed by F. M. Rites, is in quite extensive use on high-speed engines, as well as replacing the Corliss pendulum governor. It is a form of shaft governor, in which a long bar, heavy at the ends, is used instead of the weights. This bar acts by inertia and centrifugal force; that is, as the speed of the wheel increases the inertia of the bar tends to make it fall behind.

Valve Gears.

The term *valve gear* is applied to all that part of the mechanism which controls admission and discharge of steam to and from the cylinder. It includes eccentric, eccentric rod, valve-stem, valve-stem guide and valve (or valves); in automatic cut-off engines, also the mechanism by means of which the governor controls the cut-off; and in reversible engines, also the mechanism by means of which the valve motion is changed in such a way as to produce reversal of the rotary motion of the shaft.

Balanced Valves.—By means of the various forms of shaft governors, it is possible, by changing the throw and the angular advance of the eccentric, to control automatically, to some extent, the cut-off of the plain slide-valve engine. However, the live steam in the steam-chest pressing the valve against the valve-seat causes a large amount of friction. This not only absorbs a con-

siderable percentage of the work developed by the engine, but also makes the action of the shaft governor uncertain, especially for large engines. Many devices for balancing the valve, i. e., counteracting or removing this pressure from it, have been introduced, the most important being the *piston-valve* and the *flat-pressure valve*. They are all liable to give trouble from leakage, and are more or less expensive.

Riding Cut-Off Valve Gear.—It is not possible to design the common slide-valve for short cut-off, without getting bad proportions of compression, release and lead; furthermore it is difficult to make the cut-off variable while running. When it is desired to have such cut-off as $^1/_3$, $^1/_4$, $^1/_5$, etc., and to be able

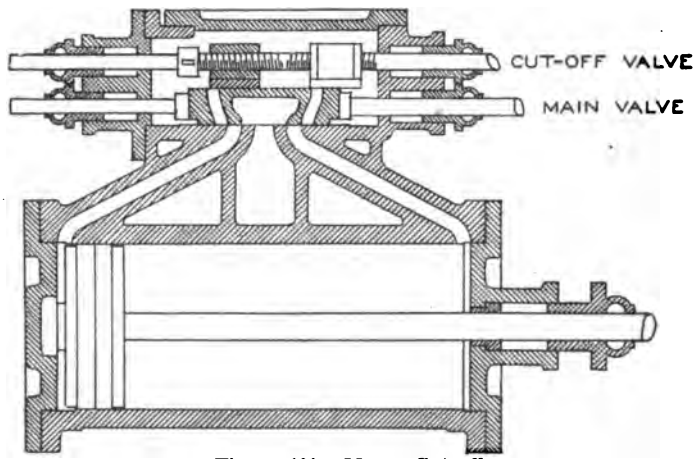


Figure 101. Meyer Cut-off.

to change it during running, an additional valve must be used. This is called the expansion valve or cut-off valve, which moves on the back of the main slide-valve, and is therefore in a way riding on it. The principle is applied in the Meyer and Rider cut-off valves.

Meyer Cut-Off.—The Meyer cut-off is illustrated by Figure 101. For each valve there must be one eccentric set at a different angle, the main valve acting like an ordinary slide-valve, while the two expansion slides can be adjusted by hand from the outside by operating a hand-wheel fixed to the valve-rod, which has right and left threads and passes through two nuts in

the expansion slides, moving the slides apart or bringing them together. On account of considerable friction, this valve is not well adapted to automatic adjusting by a governor.

Rider Cut-Off.—The Rider cut-off is in principle the same as the Meyer cut-off. It has a main slide-valve, and the back of it is formed like a cylinder with the center in the line of the slide-valve rod. In this cylindrical recess is placed the cylindrical expansion slide, which can be turned around the axis by its valve-rod, allowing more or less steam opening as it passes the oblique ports of the main valve. This expansion slide is turned by a connection from the governor.

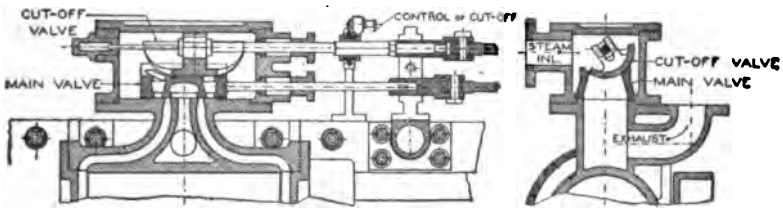


Figure 102. Rider Cut-off.

Reversing Valve Gear.—This valve gear is employed for reversing the direction of the engine's motion. In some cases this is effected by a single eccentric, in others, for instance the *link motion*, by two eccentrics, in still others by means of a loose eccentric which revolves on the shaft but is prevented from making a complete revolution by two stops. These are so placed that one arrests the eccentric in the proper position for the forward, the other one for the backward motion.

Releasing Valve Motion.—By this arrangement, the valve is liberated from the control of its moving agent and made to close by the action of a spring, weight or other force, independent of that which opened it. The Corliss and Poppet-valve engines have such an arrangement.

Corliss Valve Gear.—Figure 103 shows the valve motion of a Corliss engine.

The various makes of Corliss engines differ in design of details, but the following characteristic features are common to all of them:

(1) *Four cylindrical valves*, operated independently of

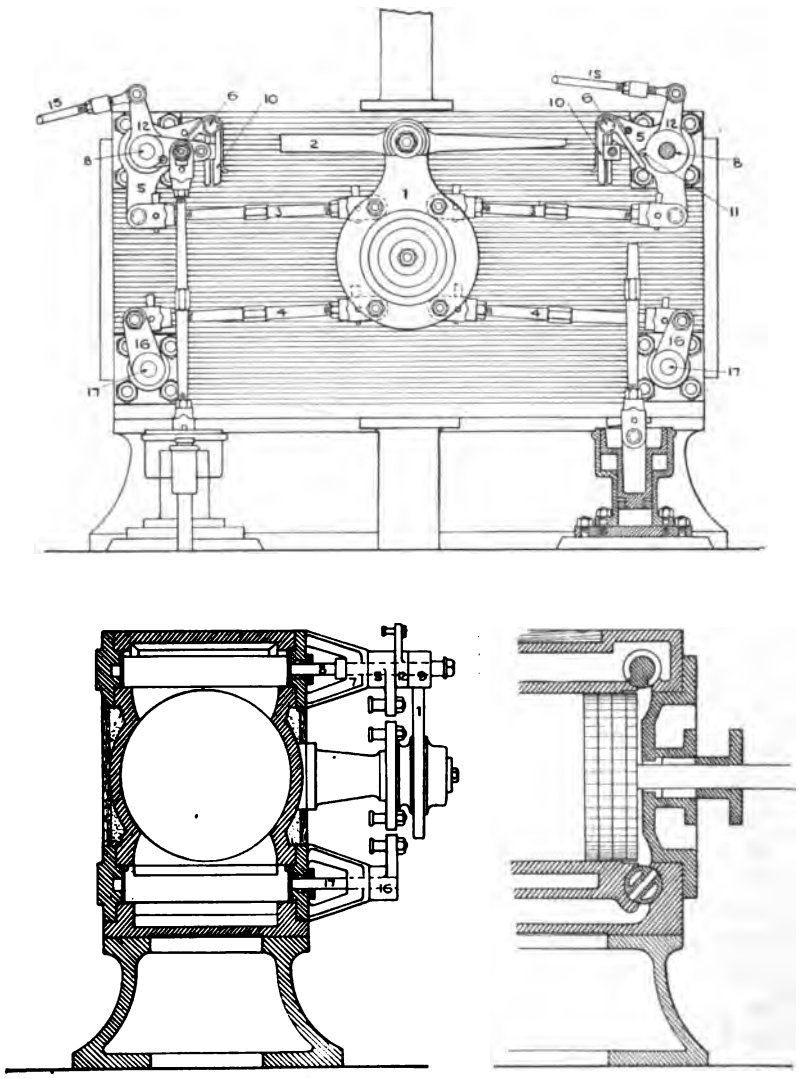


Figure 103. Corliss Valve Motion.

each other. There are two admission or steam valves with their ports, one at each end of the cylinder, and also an exhaust valve and port at each end. These valves have a rotary motion through a small arc, and are placed transversely across the cylinder, extending nearly the entire width, as indicated in Figure 103.

(2) *The wrist-plate,* an oscillating disc, which receives its motion from the eccentric and transmits it to the four valves.

(3) *The releasing mechanism,* which disengages the admission valve from its wrist-plate rod, when cut-off is wanted.

(4) *The closing apparatus,* which immediately upon release closes the admission valve.

The action is as follows:

The *wrist-plate 1* is made to rock by the *wrist-plate rod 2,* which takes its motion from a *rocker-arm* (or carrier-arm) that in turn is driven, through the eccentric-rod, from the eccentric on the shaft. The wrist-plate transmits the motion to the admission valves through the *steam-rods 3,* and to the exhaust-valves through the *exhaust-rods 4.* All of these rods are adjustable links. The *bell-crank 5,* on which is the *hook-stud 6,* turns on the *bracket 7,* through which the *valve-stem 8* passes loosely. The *steam-arm 9* is keyed to the valve-stem. When one of the steam-rods pulls towards the center of the cylinder, the bell-crank turns, lifting the hook-stud, on which the *hook 10* turns. As the hook rises, it engages the steam-arm, thereby turning the valve and admitting steam. As the bell-crank carries the hook farther, the other end of the hook strikes against a projection, the *trip* or *knock-off cam 11,* on the *knock-off lever 12,* the hook turns on the hook-stud and disengages the steam-arm, which is immediately pulled downward by the *dash-pot rod 13,* and a quick cut-off of the steam is accomplished. The action of the dash-pot is this: While the hook rises, taking the steam-arm along, the *dash-pot piston 14 is* also lifted, and a partial vacuum is created below the piston; therefore the weight of the piston and the atmospheric pressure quickly pull the steam-arm down, thus closing the steam-valve, as soon as it is released from the hook 10. The knock-off lever 12 turns on the valve-stem, and is controlled by the governor through the *governor-rod 15.* A slight motion of this lever will change the place of the trip or knock-off cam, thereby causing earlier or later disengagement of the steam-arm from the hook, and consequently earlier or later cut-off.

The exhaust rods 4 have direct, positive connection with the exhaust-valves through the *cranks 16,* which are keyed to the *valve stems 17.* The motions of these valves are not controlled

Sizes and Dimensions of Standard Corliss Engines.

SIZE OF CYLIND'R.		No. of Revolutions.	INDICATED HORSE POWER.									FLY WHEEL.			CRANK SHAFT.				STEAM PIPES.		Approx. Shipping Weight.
			80 lbs. Pressure.			90 lbs. Pressure.			100 lbs. Pressure.												
Diameter.	Stroke.		1-5 Cut-off.	1-4 Cut-off.	1-3 Cut-off.	1-5 Cut-off.	1-4 Cut-off.	1-3 Cut-off.	1-5 Cut-off.	1-4 Cut-off	1-3 Cut-off.	Diameter.	Face.	Weight in Pounds.	Diameter.	Length.	Center Shaft above Found.	Center Shaft to back of Cyl'd'r Head.	Diam. Steam.	Diam. Exh'st.	Pounds.
Inch.	Inch.											Ft.	In.		In.	Ft.	In.	Ft.	In.	In.	In.	
10	24	90	30	36	43	34	41	48	39	49	60	8	13	3500	5	6	25	10	11	3	4	10700
10	30	90	37	45	53	43	51	60	48	58	68	8	13	4300	5	6	25	13		3	4	11750
12	30	90	54	65	76	62	74	87	69	83	97	9	15	5700	6	6½	25	13	4	3½	4½	15550
12	36	85	61	78	86	70	84	98	78	94	110	10	17	6300	6	7	25	15	5	3½	4½	16900
14	36	85	83	100	118	95	114	134	107	128	150	10	19	8900	7	7½	25½	15	6	4	5	22300
14	42	82	93	112	133	107	128	140	120	144	168	12	19	8900	7	7½	25½	17	7	4	6	23700
16	36	82	105	126	140	120	144	167	135	162	187	12	21	10500	8	8	27	15	9	4½	6	26610
16	42	78	116	139	163	133	159	188	150	179	210	12	23	11800	8	8	27	17	10	4½	6	28980
18	36	80	129	155	182	148	177	208	166	199	233	12	25	12600	9	9	30	15	11	5	7	30000
18	42	78	147	176	208	168	202	237	189	227	266	14	25	13300	9	9	30	18		5	7	33800
18	48	76	162	194	227	185	222	259	208	249	291	15	25	15300	9	9	30	20	1	5	7	39570
20	42	75	175	210	246	200	240	280	225	270	314	15	29	16600	10	9	30	18	4	6	8	43450
20	48	72	192	230	268	219	263	307	246	296	346	16	29	18700	10	9	30	20	5	6	8	47612
22	42	75	211	254	296	242	290	340	271	326	382	16	31	21000	11	10	32	18	5	6	8	50600
22	48	72	232	278	308	265	318	368	298	358	410	16	31	23100	11	10	32	20	6	6	8	56220
24	48	70	268	322	380	307	368	432	345	414	486	18	33	24400	12	11	32	20	11	7	9	65140
24	60	65	311	374	438	356	427	500	401	481	561	18	37	30500	12	11	32	25	2	7	9	75740
26	48	70	315	378	445	360	432	507	405	486	570	18	37	29000	13	11½	32	21		7	9	77210
26	60	65	366	439	512	418	502	586	470	564	658	18	37	31000	13	11½	32	25	3	7	10	80000
28	48	68	355	426	508	406	487	568	457	548	680	18	37	31000	14	12	34	21	3	8	12	77980
28	54	68	400	477	554	461	546	631	523	616	709	18	37	31000	14	12	34	23	4	8	12	83000
28	60	65	424	509	594	485	582	655	545	654	763	18	37	31000	14	12	34	25	6	8	12	89000
30	48	68	407	489	571	466	559	653	524	629	734	22	46	33000	15	12½	36	21	8	8	12	93000
30	60	62	464	557	650	531	637	743	597	717	837	21	52	38500	15	12½	36	25	11	8	12	104000
30	72	55	494	593	693	565	678	791	635	762	890	24	60	52000	15	13	36	30	2			127000
32	48	65	443	532	622	507	608	710	570	684	800	24	48	34500	16	13	36	21	11			
32	60	62	529	634	739	604	725	846	680	816	952	24	60	43900	16	13	36	26	2			
32	72	55	563	675	787	643	772	902	723	868	1013	24	60	58900	16	14	36	30	5			

by the governor, and compression and release can be regulated by adjusting the lengths of the rods.

The value of a short, sharp cut-off will be shown later under Indicator Diagrams. In this respect the Corliss valve-gear gives better satisfaction than any of the automatic cut-off valve-gears previously mentioned. Another advantage is, that as there are four independent valves, each of the events of the stroke, admission, cut-off, release and compression, may be regulated as desired. Furthermore the steam-ports are very short, giving small clearance. Finally, exhaust-ports and steam-ports are separated, while in the slide-valve engine the same port that has been used for letting out the exhaust has been cooled down to the temperature of the exhaust, and is afterwards used by the entering steam, which is of much higher temperature, and therefore considerable condensation takes place. While the gearing of a Corliss engine is expensive and requires intelligent attention on the part of the engineer, the higher steam economy will easily pay for this in engine sizes of 50 to 60 horse-power and upwards. Corliss engines with the trip-release cannot be run at higher speeds than 50 to 75 R. P. M. for the larger sizes, up to 100 R. P. M. for the smallest sizes. By using a positive release instead of the trip-release, speeds up to 150 R. P. M. are made possible.

Poppet-Valve.—Engines of the poppet-valve type, which are much in use in Europe, and to some extent in this country, have a similar quick-acting drop cut-off, as the Corliss engines. The Lentz engine described below is of this type.

Lentz Engine.—This engine which is built in this country by the Erie City Iron Works, uses a double-seat poppet valve, the steam regulation being accomplished by varying the cut-off through a novel type of shaft-governor and straight-slot eccentric. Figure 104 shows a sectional view of this engine.

The cylinder has four double-seat poppet valves, one at each corner. The valve-stem has no packing, but is kept tight by a series of turned rings, called water rings. The valve-stem is actuated by an oscillating cam acting on a roller attached to the valve-stem. The cam curve is so designed as to disengage from the roller when the valve comes to a seat, but remains in contact until the valve is seated, thus preventing noise and permitting any engine speed. The cam surface is on an arm of a bellcrank, the

other arm being actuated by the eccentric rod. The eccentric shaft runs longitudinally along the outside of the cylinder, and gears with the main shaft through a special form of bevel-gear.

The action of the governor is as follows: A one-piece carrier having three arms is keyed to the eccentric shaft. A heavy inertia ring mounted on a hollow shaft, which turns freely on the eccentric shaft, gets its rotary motion through a flat circular spring. Attached to the hollow shaft is an arm carrying a pin. As the engine changes load and consequently speed, the inertia ring acts instantly, and the centrifugal weights follow as soon as the frictional resistance of the moving parts is overcome, to move the pin and the eccentric sheave along the slide block. The gov-

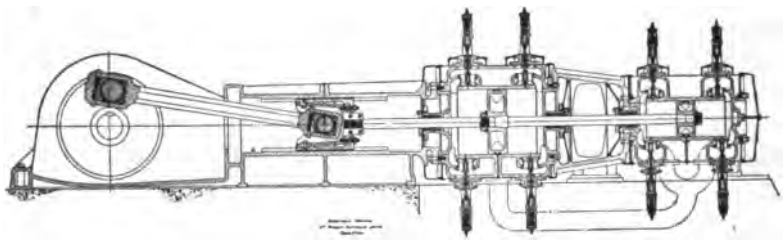


Figure 104. Lentz Engine.

ernor weights swing on pivots, and through links are directly connected to the inertia ring.

Setting Valves of Corliss Engine.

The valves of a Corliss engine are set as follows: Take off valve caps and find marks on valves and cylinder. Set the wrist-plate on center mark. Adjust valve-rods on both ends of cylinder, then move wrist-plate to outer mark and see that there is clearance between the hooking steels, and that the dash-pots are at the bottom. Proceed likewise at the other end. Now hook in wrist-plate and set the engine on dead-center; move the eccentric in the direction in which the engine is to run until steam-valve opens 1-16 inch; if the cylinder has a diameter of more than 26 inches, give 1/8 inch lead. For low-pressure cylinder give double the above amounts. After this, block up the governor 3/4 inch above safety block, move wrist-plate to outer marks and see if dash-pots drop. If they do not, adjust rods so that they will drop when the wrist-plate reaches the mark.

Corliss engines with cylinders of from 9 to 28 inches diameter should have ¼ inch lap on steam valves, and 1-64 inch exhaust openings; with cylinders from 30 to 44 inches diameter, 5-16 inch on steam valves, and 1-32 inch exhaust opening; and with cylinders from 48 to 60 inches, ⅜ inch lap on steam valves and 1-16 inch exhaust openings.

This setting is for single-ported and single-eccentric engines. For double-ported engines the setting is the same, with the exception that the lap and lead are one-half of that given above.

Engine Horse-power.

The definition of horse-power is given on page 23. In speaking of the horse-power of a steam engine, there are two kinds to be considered. The *indicated horse-power* is that calculated from the M. E. P. as given by the indicator diagram, and the *net* or *brake horse-power* is the power delivered through the shaft to the driven machinery, and is measured by a dynamometer or Prony brake.

Calculation of Indicated Horse-Power.—The horse-power of an engine can be calculated, if the area of the cylinder, the mean-effective steam pressure and the number of feet the piston travels in one minute, are known. The formula follows:

$$\text{H. P.} = \frac{P \times L \times A \times N}{33{,}000}$$

H. P. = horse-power.

P = mean effective pressure in pounds per square inch.

L = length of stroke in feet.

A = area of cylinder cross-section in square inches.

N = number of strokes per minute, or twice the number of revolutions.

Since P × A is the total average effective pressure in pounds on the piston, while L × N is feet of piston travel per minute, it follows that the product of P, A, L and N must be foot-pounds of work per minute done by the engine. Since one horse-power equals 33,000 foot-pounds per minute, it follows that the number of foot-pounds per minute divided by 33,000, gives the horse-power developed by the engine.

Example.—An engine has a cylinder diameter of 14 inches, a stroke of 36 inches and makes 80 revolutions per minute, with

a mean effective steam pressure of 45 pounds. How many horse-power does the engine develop?

Solution.—H. P. =

$$\frac{45 \times 36 \times 14 \times 14 \times 0.7854 \times 80 \times 2}{33{,}000 \times 12} = 100 \text{ approximately.}$$

For very close calculations, ½ of the area of the piston rod must be deducted from the area of the cylinder cross-section.

For the calculation of horse-power, tables of so-called *horse-power constants* or *engine constants* may also be used.

Calculation of Mean Effective Pressure.

By mean effective pressure is understood the absolute average pressure acting on the piston minus the absolute back pressure. The usual method of calculating the mean effective pressure (M. E. P.) is from the indicator diagram, which method is described in the chapter "Indicator Diagrams," page 362.

The following two methods may also be used for calculating the mean effective pressure.

Method 1.—The mean effective pressure of an engine can be calculated, if the point of cut-off, the initial pressure and the back pressure are known, by the following formula:

$$\text{M. E. P.} = \frac{\text{P} \times (1 + \text{hyp. log. R})}{\text{R}} - \text{p.}$$

P = absolute initial pressure to the point of cut-off.

R = ratio of expansion, which is the reciprocal of the cut-off, or may be found by dividing the absolute initial pressure by the absolute terminal pressure.

p = absolute back pressure.

Example.—An engine receives steam at 90 pounds gauge pressure and the steam is cut off at ¼ of the stroke. The back pressure is 16 pounds absolute. What is the mean effective pressure on the piston?

Solution.—Since the steam is cut off at ¼ of the stroke, the ratio of expansion R is 4, i. e., the steam expands 4 times. From the table on page 835 we find that the hyperbolic logarithm of 4 is 1.3863, hence we have:

$$\text{M. E. P.} = \frac{(90 + 14.7) \times (1 + 1.3863)}{4} - 16 = 62.46 - 16$$

or 46.46 pounds.

Method 2.—The mean effective pressure of an engine can also be found from the terminal pressure, the back pressure, and the ratio of expansion according to the following formula:

$$\text{M. E. P.} = P_1 \times (1 + \text{hyp. log. R}) - p.$$

P_1 = absolute terminal pressure.
R = ratio of expansion.
p = absolute back pressure.

Example.—In an engine, the steam is cut off at ¼ of the stroke and discharged at 25 pounds absolute. The back-pressure is 16 pounds absolute. What is the mean effective pressure?

Solution.—The ratio of expansion is 4. Hence:

$$\text{M. E. P.} = 25 \times (1 + \text{hyp. log. } 4) - 16.$$
$$= 25 \times (1 + 1.3863) - 16.$$
$$= 59.66 - 16 = 43.66 \text{ pounds gauge.}$$

Efficiency of Steam Engines.

When considering the efficiency of steam engines, reference may be had to either of two kinds, i. e., *mechanical efficiency* and *thermal efficiency.*

Mechanical Efficiency.—The mechanical efficiency of an engine is the ratio of the net horse-power, calculated from actual work done, to the indicated horse-power, calculated from the indicator diagram (see page 329). When the mechanical efficiency of a steam engine is given as 90 per cent, it means, that of the horse-power developed in the cylinder, as found by indicator diagram, 90 per cent is available for actual work, while 10 per cent is used for moving the engine itself (overcoming friction). The mechanical efficiency of good medium-sized steam engines at full load is 80 to 95 per cent, depending on type and working conditions. When the load drops below the normal, the mechanical efficiency also drops, but in proportion not as fast as the load. For instance, if the load is half of the normal, the mechanical efficiency drops only from say 90 per cent to about 80 per cent.

Thermal Efficiency.—The efficiency of an engine is also frequently expressed as the ratio of the heat units actually used by the engine per indicated horse-power hour, to the heat units theoretically required. The number of B. T. U. chargeable to the engine is the difference between the feed-water temperature and the boiler pressure. The following formula expresses the thermal efficiency in per cent:

$$E = \frac{W \times 100}{H}$$

E = thermal efficiency in per cent.
W = B. T. U. per hour corresponding to indicated horse-power.
H = B. T. U. in the pounds of steam consumed per I. H. P. hour above feed-water temperature.

Example.—What is the thermal efficiency of a non-condensing engine using 30 pounds of steam of 100 pounds gauge pressure per I. H. P. hour, the feed-water temperature being 212° F.?

Solution.—The indicated work done per hour in foot-pounds for each I. H. P. is 33,000 × 60. This divided by 778 gives the B. T. U. per hour. From the steam table, page 252, the total heat above 212° F. in one pound of steam of 100 pounds pressure is 1008.8 B. T. U. Substituting these values, we obtain:

$$E = \frac{33{,}000 \times 60 \times 100}{778 \times 1008.8 \times 30} = 8.41\%.$$

It will be perceived from this that the thermal efficiency of steam engines is very low.

Instead of basing the efficiency on the indicated horse-power, the brake horse-power may be used.

If we desire to express the *total thermal efficiency of the plant,* we have to calculate the ratio between the useful work done by the engine and the amount of energy furnished by the coal. In this case, the efficiency will be smaller, since there is already a considerable loss of energy in the boiler, amounting under favorable conditions to about 30%, giving a boiler efficiency of 70%.

In the above example, the thermal efficiency of the engine was 8.41%. Assuming a boiler efficiency of 70%, the total thermal efficiency of the plant will be 8.41 × 0.70 or 5.89%. This means that of every 100 B. T. U. generated by the coal, only 5.89 are transformed into useful work by the engine, the other 94.11 B. T. U. being lost.

In a compound engine, using about 15 pounds of steam per horse-power per hour, the total thermal efficiency of the plant would be brought up to 11.78%.

Maximum Efficiency.—By the term maximum efficiency, we understand the efficiency of an ideal engine working under ideal conditions. It has been proven scientifically that when a

steam engine, or any other heat engine, receives its working medium at $T_1°$ absolute, and gives it off at T° absolute, only a certain percentage of the total energy contained in the working medium at the initial temperature (T_1) can be transformed into work, even under the most ideal conditions. This percentage is expressed in the following formula:

$$E = \frac{(T_1 - T) \times 100}{T_1}$$

E = maximum efficiency in per cent.

T_1 = absolute initial temperature at which heat is furnished the engine (absolute temperature in boiler).

T = absolute temperature of heat leaving engine (absolute temperature of exhaust).

From the formula, it can be seen that the percentage of energy that can be transformed into work is greater the greater the difference between the initial and final temperatures. If all the heat were utilized in the engine and the temperature reduced to absolute zero, T in the above formula would be 0 and the efficiency would be 100%. Under practical conditions the efficiency will always be considerably below that expressed in the formula.

Example.—The boiler pressure is 100 pounds gauge, and steam of this pressure is delivered to the engine without loss. The engine exhausts at atmospheric pressure. What is the maximum thermal efficiency?

Solution.—$T_1 = 338 + 460 = 798.$

$T = 212 + 460 = 672.$

$$E = \frac{798 - 672}{798} \times 100 = 15.8\%.$$

This means, that of the energy contained in the boiler steam of 100 pounds pressure and delivered to the engine, less than 15.8% is transformed into work, even under the best possible practical conditions, as long as the steam cannot be given off at a lower temperature than 212° F. If we use a condenser, in which the exhaust pressure is brought down to 1 pound absolute, the steam is given off at 102° F. (562° absolute), and then:

$$E = \frac{798 - 562}{798} \times 100 = 29.6\%.$$

Under these conditions a higher percentage of the energy in the boiler steam may be transformed into work.

Cylinder Condensation.

When saturated steam enters the cylinder, the walls of which are lower in temperature than the steam, partial condensation takes place. Up to the point of cut-off, the steam thus condensed is replaced by steam from the boiler. After cut-off, the pressure decreases, but condensation continues until towards the latter part of the stroke, the temperature of the steam becomes lower than that of the surrounding metal, and re-evaporation occurs. This re-evaporation continues after the exhaust has opened. When steam is now admitted again to the cylinder, the temperature of the metal surface having been reduced to that of the exhaust steam, condensation and re-evaporation again take place, the process being continuous.

The steam re-evaporated during expansion does work behind the piston, but that re-evaporated during exhaust is, in the case of simple (single-cylinder) engines, lost to the atmosphere.

The amount of steam condensed varies from 10 per cent to as high as 50 per cent of the total steam.

Condensation is greater in condensing engines than in non-condensing; and is also greater the more the difference between initial and terminal pressures. (See Compound Engines, page 346). It is less for high speed than for low speed.

Besides these conditions, there are various factors reducing cylinder condensation, such as employing dry steam or superheated steam, or steam-jacketing the cylinder.

Condensing Engines.

A condensing engine is an engine in which the exhaust steam is condensed to water in an air-tight vessel. A vacuum being thus created, the resisting pressure of the atmosphere (14.7 pounds per square inch at sea level) is partly removed, and the power of the engine increased accordingly. The engine requires, therefore, a smaller quantity of steam than a non-condensing engine, to produce the same power, from 20 to 25% less steam being employed.

The various steam condensers used on condensing engines are described in the following paragraphs.

Steam Condensers.

There are two main classes of steam condensers, viz., *surface condensers* and *jet condensers*. Types of surface condensers are the *water-cooled*, the *air-cooled* and the *evaporative* condenser. Special types of jet condensers are the *ejector condenser* and the *barometric* or *waterleg condenser*.

As indicated by the names, surface condensers are those in which the steam is condensed by contact with a cool metal surface, whereas jet condensers are those in which the steam is condensed by contact with a jet of cold water.

The selection of the proper kind of condenser for a plant depends upon a great many local conditions, a few of which are: Available space; temperature and quality of water to be used for condensation purposes; available quantity of water; whether the cooling water is to be used subsequently for washing or other purposes; whether or not the condensed water can be run back into the boiler; cost of installation. All condensers, and especially jet condensers, should be provided with *safety devices* in order to prevent the water from backing into the cylinder. A tee with a back-pressure valve should be inserted in the exhaust pipe to lead off the exhaust automatically, in case it is not condensed fast enough. Furthermore, a valve should be inserted between this tee and the condenser, so that the latter can be shut off from the engine, and exhaust into the atmosphere. This will permit repairing the condenser, without shutting down the engine.

Surface Condensers.

Air-Cooled Surface Condenser.—This type is used where water is not practically available. The air may be circulated by natural or by forced draft.

Water-Cooled Surface Condenser.—This type is the most common of the surface condensers. It is a vessel generally of cylindrical form with a double head at each end. The space between the inner heads is filled with brass tubes varying in size in different constructions from ½ to 1 inch in diameter. Figure 105 shows the construction of one type of water-cooled surface condenser.

In some of these condensers, the tubes are fastened in one tube sheet, and made secure in the other by means of stuffing box

and gland. Others are constructed with stuffing box and gland at both ends, while some are rolled tight at each end. In the latter case, however, the tubes are corrugated to allow for expansion and contraction due to the different temperatures to which they are subjected. The steam usually passes through the space surrounding the tubes, and the water is forced through the tubes, though this is sometimes reversed, the steam passing through the tubes and the water through the surrounding space.

The exhaust steam of the engine, coming in contact with the cool surface of the tubes, is condensed and falls to the bottom of the apparatus, causing a partial vacuum. From here it is removed by means of an air-pump, which at the same time performs

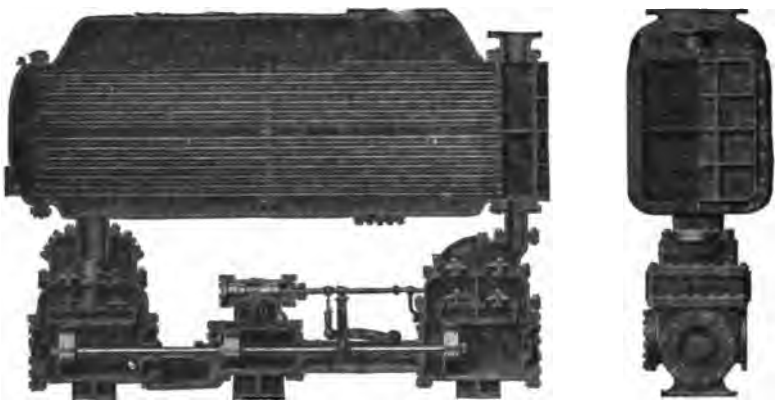


Figure 105. Wheeler Surface Condenser.

the duty of removing the air that exhaust steam always contains. The air and condensed steam may also be removed by separate pumps. The circulation of water in the tubes is usually kept up to a normal amount per minute by a circulating pump, or by letting the cooling water flow from some higher point through the condenser by gravity. More cooling water is required than in any other type of condenser.

Tube Surface.—The number of square feet of cooling surface required must be based primarily on the amount of heat to be abstracted from the exhaust steam. The amount of surface varies from about 1 to 3 square feet per indicated horse-power. This depends on a variety of conditions, such as temperature of

cooling water, temperature of exhaust, amount of steam used per I. H. P., and the coefficient of heat transmission. This coefficient again is dependent on a number of conditions, and varies considerably in value. For brass, under ordinary conditions, with the old-style surface condenser and with a water velocity of from 50 to 100 feet per minute, it amounts on the average to 250. With velocity of water of 100 to 240 feet per minute, it is about 50% greater. With the modern dry-tube surface condenser, where the water of condensation is removed rapidly, higher values are used, an average being 600, with a water velocity of 240 to 300 feet per minute. The value in all cases increase approximately as the square root of the velocity of the water in the tubes.

The following formula may be used for calculating the surface:

$$S = \frac{W \times (T - T_1)}{C \times \left[T_2 - \left(\frac{t + t_1}{2} \right) \right]}$$

S = condenser tube surface in square feet.
W = weight of steam in pounds condensed per hour.
T = total heat above 32° F. in one pound of exhaust steam of condenser pressure.
T_1 = total heat in one pound of condensed steam above 32° F. at hot-well temperature, equals temperature minus 32.
T_2 = temperature of exhaust steam at condenser pressure.
t = initial temperature of circulating water.
t_1 = final temperature of circulating water.
C = coefficient of heat transmission in B. T. U. per hour per degree difference in temperature of exhaust steam and average circulating water, per square foot of surface. It may vary in value from about 250 to 800.

The formula may easily be worked out, from the fact that the total heat abstracted from the exhaust steam must be equal to the total heat absorbed by the cooling water. In this formula, it is assumed that the exhaust steam is in a dry saturated state. In practice this is not true, so that for a more accurate result, the total heat in the steam (T in the formula) should be multiplied by the quality of the steam. If not known, the quality of the steam may be taken at about 85%.

Example.—A 375 H. P. engine uses 18 pounds of steam per I. H. P. hour. The temperature of the cooling water is 60° F., and the vacuum is 26 inches. In an old-style surface condenser, what should be the tube-surface?

Solution.—

$W = 375 \times 18 = 6{,}750$ pounds.

$T = 1{,}114.2$, as obtained from the steam table, page 252.

$C = 250$.

$T_1 = 124.3 - 10 - 32 = 82.3$. From the steam table, 124.3 is the temperature corresponding to 26 inches vacuum. From this 10 is subtracted, as this may be considered the average difference between the temperature of the hot-well and that corresponding to the degrees of vacuum (for a multi-flow condenser, about 5 should be subtracted).

$T_2 = 124.3$ from steam table.

$t = 60$.

$t_1 = 124.3 - 25 = 99.3$. Average practice shows that about 25 must be subtracted from the temperature corresponding to the degree of vacuum (for a multi-flow condenser, about 10 should be substracted).

$$S = \frac{6{,}750 \times (1{,}114.2 - 82.3)}{250 \times \left(124.3 - \frac{60 + 99.3}{2}\right)}$$

$$= \frac{6{,}750 \times 1{,}031.9}{250 \times 44.6} = 625 \text{ square feet.}$$

This corresponds to $\frac{625}{375} = 1.67$ square feet per I. H. P. hour.

Cooling Water for Surface Condenser.—The method of calculating the cooling water required for a surface condenser is similar to that for a jet condenser (page 340). The formula is

$$W = \frac{h}{t_1 - t}$$

W = pounds of cooling water required per pound of steam.

h = B. T. U. given up by one pound of steam in condensing to water of hot-well temperature.

t_1 = final temperature of cooling water in degrees F.

t = initial temperature of cooling water in degrees F.

Evaporative Condenser.—Where the supply of condensing water is very limited, the so-called *evaporative condenser* is often adopted. The efficiency of this kind of condenser has not received the recognition among engineers that it deserves.

This condenser consists mainly of a battery of copper, brass or iron pipes, affording a large radiating surface, the exhaust steam passing through the tubes, and the condensing water trickling over the surface. The passage of air over the thin film of water removes the vapor as it rises, and thus hastens evaporation caused by the steam in the pipes, the steam being condensed at the same rate as it gives off heat. The *amount of cooling water* required with this kind of condenser has been shown by experiment to be about equal to the weight of the steam used in the engine.

The surface of the condenser can be greatly reduced if the velocity of the current of air is increased by mechanical means, for instance by a fan. In case a fan is employed, only about one-third of the surface that would otherwise be necessary is required. Without forced draft, and if the engine exhausts the steam at 140° F., one square foot of surface evaporates 0.8 of a pound of water per hour. If, however, a fan is used, one square foot can evaporate 2.4 pounds of water per hour. Therefore a fan in this case increases the capacity of a condenser three times.

The atmospheric ammonia condenser illustrated in the chapter "Refrigeration" shows the construction of the above steam condenser.

The Jet Condenser.

Construction.—The jet condenser is less bulky than the surface condenser. In this condenser, the steam from the exhaust pipe of the engine meets a spray of cold water, is condensed and then removed with the condensing water by a wet-air pump. The latter must be larger than the one used on the surface condenser, as it not only has to remove the air and the condenser water, but also the circulating water. Instead of employing only one pump, the liquid may be removed by one pump, and the air by a separate pump, or by means of a barometric pipe and a dry vacuum pump. The construction of a jet condenser is shown in Figure 106.

When designing a jet condenser, the main consideration is that there should be no possibility of the condensed steam and injection water passing back into the engine. To effect this it is customary to have the exhaust steam enter at the top, while the water falls to the bottom where it is removed by a pump or by a barometric pipe.

The volume of the condenser should be about one-half to one-fourth of that of the steam cylinder or cylinders directly exhausting into it.

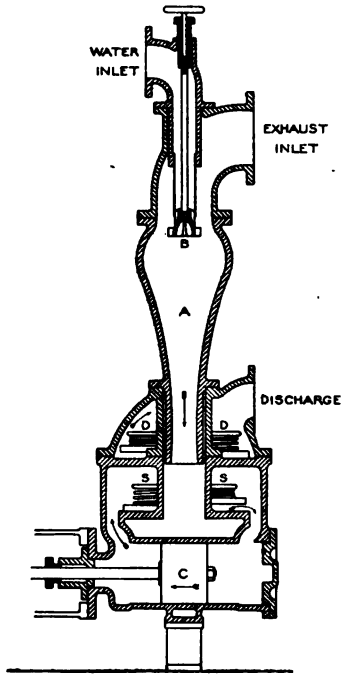


Figure 106. Jet Condenser.

Quantity of Cooling Water Required.—In practice with reciprocating engines, the temperature of the hot-well, i. e. the hot water leaving the condenser, is about 110 to 120° F. and in extreme cases as high as 130°. F.

To find the quantity of *cooling water per pound of steam* for a jet condenser, the following formula may be applied, though in practice from 5 to 15% must be added.

$$W = \frac{h \times p}{D}$$

W = number of pounds of cooling water required to condense one pound of steam.

h = number of B. T. U. given up by one pound of steam in condensing to water of hot-well temperature.

p = number of pounds of steam condensed.

D = difference in temperature between cooling water and hot-well.

Example.—The exhaust steam enters the condenser at 125° F., and the temperature of the hot-well is 110° F. If the cooling water enters the condenser at 80° F., how many pounds of cooling water are necessary per pound of steam?

Solution.—

$$W = \frac{1036.5 \times 1}{30} = 34.5 \text{ pounds approximately.}$$

The following remarks explain the calculation:

The number of pounds of steam condensed (p), in this case = 1.

The number of B. T. U. (h) given up by the steam in condensing to water of hot-well temperature is found as follows:

The total heat above 32° F. in one pound of steam at 125° F. (from steam table page 252), is 1,114.5 B. T. U., and the total heat in one pound of water leaving the condenser is 110 — 32 or 78 B. T. U. Consequently h = 1,114.5 — 78 or 1036.5 B. T. U., which amount must be absorbed by the cooling water.

The difference between the initial and the final temperature of the cooling water (D), or its increase in temperature due to the heat absorbed, is 110 — 80 or 30° F. This means that each pound of cooling water takes up 30 B. T. U., and to absorb the total heat (1036.5 B. T. U.), there are consequently necessary: $\frac{1036.5}{30} = 34.5$ pounds of cooling water per pound of steam. This is the theoretical value, and is furthermore based on the assumption that the steam is dry.

Quantity of Water for Condensing Engine.—The cooling water required for a certain engine may readily be calculated from the above, when the horse-power and the pounds of steam consumed per horse-power hour are known.

Example.—If a condensing engine uses 23 pounds of steam per H. P. per hour and develops 180 H. P., what will be the water consumption, in gallons per minute, of the jet condenser connected with the engine? Assume the same conditions as in the previous example, i. e., steam is discharged at 125° F. into the condenser, temperature of the hot-well is 110° F., and temperature of cooling water is 80° F., the condenser requiring 34.5 pounds of cooling water per pound of steam.

Solution.—Since 23 pounds of steam are consumed per horse-power per hour, and the engine develops 180 horse-power, the steam consumption is $23 \times 180 = 4{,}140$ pounds per hour. This would require $4{,}140 \times 34.5 = 142{,}830$ pounds of cooling water per hour $= \frac{142{,}830}{60} = 2{,}380.5$ pounds per minute. Since 1 gallon of water of 80° F. = 8.32 pounds, the number of gallons per minute required $= \frac{2{,}380.5}{8.32} = 286.1$ gallons per minute.

This may also be calculated by the following formula, which leads to the same results, except that all the steps are first indicated, thus allowing cancellation.

$$G = \frac{h \times p \times H.\,P.}{60 \times 8.32 \times D}$$

G = gallons of water per minute.
h = number of B. T. U. given up by the steam in condensing to water of hot-well temperature.
p = number of pounds of steam condensed and cooled to the hot-well temperature, per horse-power per hour.
H. P. = horse-power of engine.
D = difference in temperature between cooling water and hot-well.
8.32 = weight in pounds of one gallon of water at 80° F.

Solution.—

$$G = \frac{1036.5 \times 23 \times 180}{60 \times 8.32 \times 30} = 286.1 \text{ gallons per minute.}$$

Barometric Condenser.—This type of jet condenser, shown in Figure 107, is also called *gravity* or *water-leg condenser.* A cylindrical chamber is placed above the discharge pipe, at a distance of 34 feet from the hot-well. The steam and cooling water meet in this chamber and the steam is condensed, whereby a

partial vacuum is produced. As is well known, the atmospheric pressure supports a column of water of about 34 feet, and this condenser thus allows removing the water from the condenser by gravity without the employment of a pump. In some types, a small air-pump is also used, for drawing away the air not removed with the water.

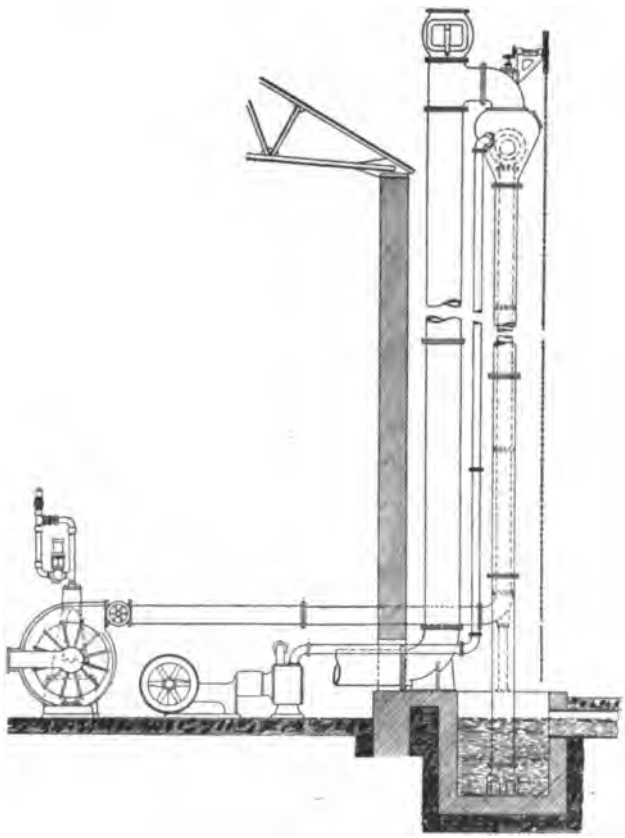


Figure 107. Barometric Condenser.

Ejector Condenser.—This type of condenser is also called the *injector condenser*. It acts on the principle of an injector, and maintains a vacuum without the use of an air-pump. It has the disadvantage that it is not as well suited to varying loads as the other condensers.

Condenser Pumps.

Condenser pumps are classified as dry-air pumps and wet-air pumps. The former handle air alone, while the latter handle both air and water.

Size of Wet-Air Pump for Jet Condenser.—To find the diameter of the air pump (double-acting) necessary for the engine considered on page 342, if the pump should have an 18-inch stroke and make 50 full strokes per minute, the following formula may be used:

$$d = \sqrt{\frac{2.75 \times H.P. \times (p_1 \times p + p_1) \times 1{,}728}{2 \times w \times n \times s \times 0.7854 \times 60}}$$

d = diameter of pump in inches.
H. P. = horse-power developed by engine.
p = pounds of cooling water per pound of steam.
p_1 = pounds of steam condensed per horse-power per hour.
w = weight in pounds of one cubic foot of water at temperature of hot-well.
n = number of single strokes of pump per minute.
s = length of one stroke in inches.

This formula may be reasoned as follows: The capacity of the pump must be equal to the amount of water used for condensing and the amount of steam condensed, times a certain factor, generally taken as 2.75. The reason for multiplying by this factor is that the pump not only has to remove the volume of water but also a large quantity of air, contained partly in the exhaust steam, partly in the condensing water, and therefore the displacement in the air-pump for a jet condenser must be in excess of the volume of water to be pumped. Generally, as stated above, this displacement in the pump is taken to be 2.75 times the volume of the water that must be handled by the pump.

Since the pump is double-acting, it makes two deliveries for each full stroke. Hence the capacity of the pump in cubic inches per minute (according to formula for calculating volume of cylinders) is: $d \times d \times 0.7854 \times 1 \times n \times 2$. To change to cubic feet per minute, it is divided by 1,728. To change to pounds of water of hot-well temperature, it is multiplied by w, the weight of a cubic foot of water of this temperature, (in this case 61.89 pounds, the temperature being 110° F.). By then multiplying by 60, the pounds of water per hour are obtained. With

these changes the formula for the capacity of the pump in pounds per hour becomes:

$$\frac{d \times d \times 0.7854 \times 1 \times n \times 2 \times w \times 60}{1{,}728}$$

The amount of water to be pumped out consists of the cooling water plus the condensed steam. In the previous problem it was found that the engine required 34.5 pounds of cooling water per pound of steam. Hence the total pounds of cooling water $= p \times p_1$, or 34.5×23. To this must be added the pounds of condensed steam $= p_1$, or 23. Thus, total water and air $= (p_1 \times p + p) \times 2.75$.

By equating these two, transposing (for method of transposing, see page 808), and solving for d, the formula given is obtained.

On substituting the various values in the formula, the following result is obtained:

$$d = \sqrt{\frac{2.75 \times 180 \times (23 \times 34.5 + 23) \times 1{,}728}{2 \times 61.89 \times 50 \times 18 \times 0.7854 \times 60}} = 11.6 \text{ inches.}$$

The diameter of the air pump must be 11.6 inches, or in other words, an 11.6×18 inch double-acting pump, making 50 full strokes per minute, must be used.

Size of Wet-Air Pump for Surface Condenser.—In this case, the pump handles condensed steam and air. Experience shows that the quantity of air and water which the pump must handle is, on the average, 10 times the volume of condensed steam, for reciprocating engines, and 20 times for steam turbines. These figures are of course subject to large fluctuations, and hence no definite rule can be given.

Size of Dry-Air Pump.—These pumps are used for removing only the non-condensable vapors. Average practice seems to show that their capacity should be such as to handle 20 to 30 times the volume of condensed steam, where the vacuum is less than 27 inches, and about 50 times, where the vacuum is 28 inches or over.

Size of Separate Pumps.—Where separate pumps are used for the water and for the air, we have for the water pump for jet condensers, the same formula as given on page 344, leaving out the factor 2.75. For surface condensers, the capacity is calculated on a basis of the amount of condensed steam. These

The latter method may be expressed by the following simple formula:

$$N = \frac{P}{p}$$

N = total ratio of expansion.
P = absolute initial pressure in high-pressure cylinder.
p = absolute terminal pressure in low-pressure cylinder.

Example.—A compound engine works with an initial pressure of 130 pounds absolute, expanding to a terminal pressure of 26 pounds absolute in the high-pressure cylinder. The steam thus received by the low-pressure cylinder is expanded to 8 pounds absolute pressure. What is the total ratio of expansion?

Solution.—

$$N = \frac{130}{8} = 16.25 \text{ expansions.}$$

Size of Cylinders of Compound Engines.—The measure for compound engines is the low-pressure cylinder. It is made large enough to carry the entire load through the range of pressure from the initial to the terminal. It must be made so large because it receives the steam at a low pressure, and the lower the pressure, the larger must be the cylinder area in order to obtain the same amount of work.

After the proper size of the low-pressure cylinder has been calculated, the pressure range is generally distributed on both the low and the high-pressure cylinders, so that the low-pressure cylinder will carry only one-half or nearly one-half of the load for which it originally had been selected. The diameter of the high-pressure cylinder can then be calculated.

The diameter of a steam cylinder is found according to this formula:

$$d = \sqrt{\frac{\text{H. P.} \times 33{,}000}{\text{M. E. P.} \times s \times 0.7854}}$$

d = diameter of cylinder in inches.
H. P. = horse-power to be developed by engine.
M. E. P. = mean effective pressure in pounds per square inch.
s = piston speed in feet per minute.

Example.—What must be the diameter of the high- and low-pressure cylinders of a compound engine of 500 H. P. capacity, the piston speed being 480 feet per minute, if the absolute initial pressure at which the engine receives the steam is 135 pounds, the absolute terminal pressure in the low-pressure cyl-

inder 9 pounds, and the steam is condensed into a vacuum of 24 inches?

Solution.—First the diameter of the *low-pressure cylinder* is calculated, this being done as though the whole load were to be carried by this cylinder only. The total ratio of expansion is $\frac{135}{9} = 15$.

The steam is condensed into a vacuum of 24 inches. Since 30 inches pressure equals 14.7 pounds, this vacuum corresponds to $30 - 24 = 6$ inches pressure or $\frac{6 \times 14.7}{30} = 2.94$ pounds, or approximately 3 pounds. This is the absolute back pressure.

From the formula given on page 330, the mean effective pressure can be calculated thus:

$$\text{M. E. P.} = \frac{135 \times (1 + \text{hyp. log. } 15)}{15} - 3.$$

From the table page 835, the hyperbolic logarithm of 15 is found to be 2.7081. Hence, M. E. P. $= 9 \times (1 + 2.7081) - 3$ $= 30.4$ pounds.

By inserting the values given, and that found by calculation, in the formula, the diameter of the low-pressure cylinder is obtained:

$$d = \sqrt{\frac{500 \times 33{,}000}{30.4 \times 480 \times 0.7854}} = 37.95 \text{ inches.}$$

A cylinder of these dimensions would be sufficiently large to produce all the power required, the steam going through 15 expansions. However, since so many expansions in one cylinder is not economical, the low-pressure cylinder is allowed to carry only half the load, the other half being given to the high-pressure cylinder.

The mean effective pressure in the low-pressure cylinder should therefore be $30.4 \div 2$ or 15.2 pounds. Since the absolute back pressure is 3 pounds, the absolute mean pressure will be $15.2 + 3$ or 18.2 pounds.

In order to be able to calculate the diameter of the *high-pressure cylinder,* its terminal pressure must be known. If there is no drop, the terminal pressure of the high-pressure cylinder is equal to the initial pressure of the low-pressure cylinder. This

can be found from the absolute terminal pressure and the absolute mean pressure of the low-pressure cylinder as follows:

From the formula given on page 331, by substituting absolute mean pressure for the M. E. P. plus absolute back pressure, we obtain:

Absolute mean pressure = absolute terminal pressure $\times$ (1 + hyp. log. R).

Transposing this formula gives:

$$\text{Hyp. log. R} = \frac{\text{Absolute mean pressure}}{\text{Absolute terminal pressure}} - 1$$

$$= \frac{18.2}{9} - 1 = 1.0222.$$

The number corresponding to this, as found from the table, page 835, is 2.78, and consequently the ratio of expansion is 2.78. The initial pressure of the low-pressure cylinder and consequently also the terminal pressure of the high-pressure cylinder is, therefore, 2.78×9 or 25 pounds absolute.

The ratio of expansion in the high-pressure cylinder is $135 \div 25$ or 5.4, the hyp. log. of which is 1.6864. Consequently, the mean effective pressure in the cylinder must be:

M. E. P. $= 25 \times (1 + 1.6864) - 25 = 42.16$ pounds, and the diameter of the high-pressure cylinder is:

$$d = \sqrt{\frac{250 \times 33{,}000}{42.16 \times 480 \times 0.7854}} \text{ or } 22.78 \text{ inches.}$$

The area of the high-pressure cylinder can also be found more directly by dividing the area of the low-pressure cylinder by the ratio of expansion in that cylinder, which, if applied to the above case, would be:

$$\frac{38 \times 38 \times 0.7854}{2.78} = 408 \text{ square inches.}$$

The diameter of the high-pressure cylinder would then be:

$$d = \sqrt{\frac{408}{0.7854}} \text{ or } 22.78 \text{ inches.}$$

The piston speed of the engine is 480 feet per minute, and if we select a stroke of, for instance, 4 feet, the engine must then make 60 revolutions per minute. The dimensions of the engine would then be approximately $22'' \times 38'' \times 48''$. The absolute initial pressure is 135 pounds, there are 15 expansions, and the

engine exhausts into a condenser against 3 pounds back pressure, which corresponds to about 24 inches of vacuum. The cylinder ratio is: $\frac{38 \times 38}{22 \times 22} = 2.98$, or about 1 to 3.

The ratio of cylinder areas of compound condensing engines is usually greater than the previous example shows. It is generally 1 to 4.5 to 1 to 5 and even as high as 1 to 7 and 1 to 8.

Receiver.

A receiver should be placed between the two cylinders of a cross-compound engine, the high-pressure cylinder exhausting into this receiver. It is usually of cylindrical shape and is provided with a steam jacket or some other means for re-heating the steam, which arrangement is called the *re-heater.*

Object of Receiver.—The object of the receiver and re-heater is two-fold. In the first place, it replaces any heat lost by radiation in the high-pressure cylinder, or after the steam has left it. Secondly, in a cross-compound engine with the cranks at an angle of 90°, the low-pressure cylinder is not ready to receive the steam at the time it is discharged from the high-pressure cylinder, therefore it is necessary to have a receiver for steam storage.

A receiver is not necessary in tandem compound engines, since loss by radiation is insignificant and the distribution of the load cannot be influenced.

Dimensions of Receiver.—The volume of the receiver is usually from 1 to 1.75 times the volume of the high-pressure cylinder, when the cranks are at 90° to 120°, the larger ratio being more favorable to a straight back pressure line of the high-pressure cylinder and steam line of the low-pressure cylinder, as can be proven by indicator cards. The angle at which the cranks are set affects the required volume of the receiver.

Receiver Pressure.—In order that the work of a compound engine may be divided equally, or nearly so, between the two cylinders, the receiver pressure should be proportioned according to the formula given below. An equal distribution of the load between the two cylinders is brought about by the adjustment of the cut-off in the low-pressure cylinder. Shortening the cut-off increases the receiver pressure and forces the

low-pressure cylinder to perform a greater portion of the work, while lengthening the cut-off has the opposite effect.

The pressure in the receiver can be calculated according to this formula:

$$P = \frac{d^2 \times p}{d^2 + D^2}$$

P = absolute receiver pressure in pounds per square inch.
p = absolute initial pressure in pounds per square inch.
d = diameter of high-pressure cylinder in inches.
D = diameter of low-pressure cylinder in inches.

Example.—What should be the receiver pressure of a compound engine with a high-pressure cylinder of 28-inch diameter and a low-pressure cylinder of 52-inch diameter, the initial absolute pressure being 140 pounds and the absolute terminal pressure in the low-pressure cylinder 7 pounds? A vacuum of 24 inches is maintained.

Solution.—

$$P = \frac{28 \times 28 \times 140}{28 \times 28 + 52 \times 52} = 31.47.$$

The receiver pressure must be 31.47 pounds absolute, or 16.77 pounds gauge pressure.

Triple Expansion Engines.

In a triple expansion engine the steam is expanded in three stages, passing through three cylinders consecutively. Triple expansion engines are costly, and the installation of them does not pay unless the steam pressure is very high, i. e. from 175 to 250 pounds. These engines, as well as *quadruple expansion* engines, are not very frequently used on land, being mostly used on ocean steamers of large capacity.

Average Steam Consumption of Corliss Engines.

Kind of Engine.	Lbs. of steam per I. H. P. per hour.
Non-condensing	28.0
Condensing	23.0
Compound non-condensing	21.0
Compound condensing	12.5
Triple expansion condensing	11.0
Quadruple expansion condensing	10.5
Multiple expansion condensing	9.5

High-speed Engines.

The continuous growth of electric light and power plants has necessitated the introduction of high-speed engines for driving the dynamos.

A high-speed engine, in order to give satisfactory results, must be of simple construction, run at high rotatory speed, be economical in the use of steam, and it must maintain as constant a speed as possible. The latter point is of great importance, since engines that are employed for driving dynamos are often subjected to sudden, and frequently considerable, variations of load.

The desired results can be obtained by means of the proper valve gear and governor, the so-called fly-wheel governor having been designed for this purpose. Figure 110 illustrates the construction of such a governor.

Figure 110. Shaft Governor.

Shaft-Governor.—In this governor, two weights are carried on arms which are pivoted to the fly-wheel, and springs are fastened to these arms and to the rim of the wheel. Links connect the arms and the collar which is placed loosely on the main shaft and carries the eccentric. When the fly-wheel revolves, the weights have a tendency to travel outward, i. e. away from the shaft, but the centrifugal force is counteracted by the tension of the springs. If the speed is increased sufficiently, this tension is overcome, the weights move outward, and thereby shift the eccentric, thus changing its eccentricity, angle of advance or both, and producing an earlier cut-off in the cylinder. On the other

hand, if the speed becomes slower the operation is reversed and a later cut-off takes place.

Governors should be kept perfectly clean, and free from accumulations caused by the use of inferior oils, as such gummy substances have a tendency to interfere with the easy movement of the different parts, thus causing unequal regulation. With the aid of a well-designed governor of the above type it is possible to regulate the engine to within one per cent of its rated speed from full load to no load or vice-versa.

Advantages of Two Valves.—The majority of high-speed engines are operated by a single valve, which has the disadvantage that if the angle of advance only is altered, the cut-off may be varied to suit the load, but then, the lead will also vary in such a way, that it will increase as the cut-off decreases. If, on the other hand, the regulation is performed by varying the eccentricity alone, the reverse takes place. For shaft regulation in connection with a single valve it therefore becomes necessary to vary the eccentricity and angle of advance simultaneously. This, however, is not the case where separate distribution and expansion valves are used, as for instance, in the *Buckeye engine,* because the admission and exhaust closures are effected by the distribution valve, while the governor acts only on the cut-off.

Steam Consumption.—Under ordinary conditions, high speed engines consume from 30 to 35 pounds of steam per indicated horse-power per hour.

Engine Foundations.

The material most used for engine foundations is concrete made of Portland cement. The proportions are 1 part of cement, 3 parts of torpedo sand and 4 parts of crushed stone. These are well mixed and stamped in. Satisfactory results are also obtained from good hard burnt brick laid in cement mortar, every course being properly wetted and grouted.

The fly-wheel pit should be proportioned for a reasonable amount of room. The walls should be washed, and plastered with first-class cement; the bottom should be treated in the same manner, and should, if possible, be connected to the sewer. Templates for foundations should be handled with care, the center lines of the cylinder and main shaft should be at right angles

with each other and all bolt holes should be very carefully marked as per foundation drawing.

The template should be well braced to preserve its alignment, and should be leveled and located from existing lines, such as building walls or existing countershafts.

The foundation bolts should be placed in the template holes. A pipe or a square wooden box about three to four inches in diameter and approximately two feet long should be slipped over the bolt and gradually raised as the foundation is built up. This leaves some space around each bolt and allows it to be slightly moved if necessary, when the engine bed plate is lowered on to the foundation. The holes are afterwards filled with a mortar, of either pure cement or of a mixture of half sand and half cement. The tops of the bolts should be placed in their proper height before the foundation is built.

The cap stone, if one is used, should be laid as level as possible, as this facilitates the work of leveling and lining the engine later on.

After the engine has been placed on the foundation, it should be lined and leveled with the greatest care. The work cannot be done too carefully, as the smooth running and the life of the engine to a great extent depend upon the accuracy in its erection.

The crank shaft is best lined up by means of a water-proof silk line drawn through the center of the cylinder and extended a convenient distance beyond the main bearing. The crank pin should be brought up to the line in nearly extreme forward and backward positions, and when the distance from line to face of crank pin hub measures alike in both positions, the shaft is at right angles with the cylinder. A level should be used to adjust the shaft horizontally. If the shaft is exactly level and exactly at right angles to the cylinder and if the guide is bored out at the same time as the cylinder is bored out, i. e. with the same boring bar, without changing its position, then no trouble need be feared for the future operation of the engine.

Steam Engine Indicator Diagrams

Examination of Engine by Means of Indicator.

The purpose of an indicator is to determine whether a steam engine uses the steam economically or not, whether the valves are set properly or not, whether the steam admission is correct and the cushion a proper one, whether the cut-off is even, or whether the piston or valves leak. It is the quickest means of finding whether or not the steam in the cylinder works properly, and has the advantage that it can be applied without stopping the engine.

Steam engines, as well as ammonia compressors and pumps, should be provided with openings for indicator, and means for attaching the indicator cord to the cross-heads.

The indicator should preferably be equipped with a reducing pulley, so as to accommodate all strokes of engines and compressors. Such a set can be had complete in a case. If the connections are permanently made to the cylinders, such cylinders may be indicated in a few minutes, and by reading the cards the engineer is enabled readily to locate many difficulties.

Description of Indicator.

The indicator is in certain respects like a steam cylinder. In a small cylinder is placed a tight-fitting piston, below which the pressure is admitted. A spring is put above the piston, and is secured to it. Several springs of different tensions are furnished with the indicator, and they can be easily interchanged. In selecting the spring to be used the object should be to obtain as high a diagram as possible, as this will be least subject to error. The speed and pressure must be considered in selecting the spring.

To this small indicator cylinder is attached a drum, which is held against the starting stop by a coiled spring, which is so made

that the pulling cord is taut at all times. At the end of the travel of the drum another stop is provided. The cord, which must be of good quality, so as not to stretch (a good fishing line will answer the purpose very well), runs over a little grooved pulley which is secured to the bottom of the drum, the cord being guided by a swivel roller. The piston has a piston rod extending through the cylinder-head and connected to a parallel motion, which carries a pin on the one lever for marking on the paper which is put on the drum. It is best to use metallic coated paper and a composition pin, as this gives the sharpest lines. The head of the cylinder is screwed on, to facilitate taking out the spring, as well as cleaning.

To a screw at the bottom of the drum is attached the reducing pulley, which carries the cord fastened to the cross-head, and has a stiff spring inside to keep the cord taut. On the shaft of this large pulley small ones can be fastened, of which a number are furnished, and which are marked for the length of stroke they can be used for.

Putting Indicator in Place.

The indicator is attached to the end of the cylinder, through holes drilled and tapped for half-inch pipe, either at the side or top, whichever is the more convenient, and connected with a reducing motion, attached to the cross-head.

The holes must be drilled in such a way that they communicate with the clearance space, and cannot become covered by the piston when the latter is at the extreme end of the cylinder.

In most engines the holes are drilled and tapped in the shop before the engine leaves the factory. Sometimes, however, this is not done, and care should then be taken, when drilling the necessary holes, to choose the proper place and to prevent the chips from the drill getting into the cylinder, especially the front end.

As the length of the diagram represents the travel of the piston for one-half revolution, or one stroke of the engine, and the length of this diagram is determined by the rotation of the paper drum, this motion must be taken from some part of the engine which has a motion coinciding with that of the piston. The cross-head is the most convenient part for this purpose.

This drum motion, or reducing motion, as it is called, may

be obtained by the Brumbo pulley, the pantograph, reducing wheels, or other devices, of which there are a large number. The important point is that they reduce the motion proportionally to the stroke of the engine and parallel with the cross-head or piston, so that, for instance, when the cross-head has made one-fourth of its stroke the pencil will have traveled one-fourth the length of the diagram.

Before taking a diagram from an engine the indicator should be thoroughly warmed by opening the steam cock between the indicator and the cylinder. The working parts should move smoothly. The spring should be so chosen as to furnish a diagram from 1½ inches to 2 inches high. Dividing the gauge steam pressure by 1¾ gives the scale of spring that will make a diagram of proper height.

The pencil should be sharp and smooth and so adjusted that when the pressure is applied it will make a smooth, fine, distinct line. The cord leading from the cross-head or reducing motion to the indicator should be so adjusted that the paper drum in rotating will not touch the stops.

After the paper has been placed taut and smooth around the drum the paper drum is set in motion and the shut-off cock opened, allowing the steam to enter below the piston. The pencil is pressed against the paper for one revolution or more, the steam then shut off to the indicator, and the pencil again pressed to the paper. This gives the atmospheric line, which is drawn with atmospheric pressure on both sides of the piston.

As many as possible of the following data should be obtained and noted on the diagram: Date, hour of taking diagram, type or make of engine, which end of cylinder and which engine of one of a pair, diameter of cylinder, length of stroke, diameter of piston rod, number of revolutions per minute, steam pressure by gauge at the boilers and at the engine, atmospheric pressure by barometer, vacuum by gauge in the condensers, if the engine is a condensing type, temperature of feed water at boilers, receiver pressure by gauge if the engine is compound, and back pressure by gauge if simple; scale of spring used in taking diagram; also any other data connected with the plant that may be of importance.

Indicator Diagrams.

The indicator diagram is the result of two motions. The horizontal motion of the paper corresponds exactly with the movement of the piston; it represents the stroke of the engine on a reduced scale. The vertical movement of the pencil is in direct proportion to the steam pressure in the cylinder. The resulting curved line consequently represents by its length the stroke of the engine, and by its height at any point, the pressure on the piston at a corresponding point in the stroke.

The shape of the diagram depends very much upon the manner in which the steam is admitted to and released from the cylinder of the engine. It gives information on the following points:

Arrangement of valves for admission of steam, cut-off, release and compression of steam.

Adequacy of steam-ports and passages for admission and exhaust, and when applied to the steam chest, adequacy of the steam pipe.

Amount of power developed, quantity lost in various ways, i. e., by wire drawing, back pressure, premature release, or any other maladjustment of the valves.

The indicator diagram has proven very useful to designers of steam engines, and the degree of excellence to which the steam engine of to-day has been brought is due largely to the use of the indicator. A careful study of the diagrams under different conditions of load, pressure, etc., is the only means of becoming familiar with the action of steam in the cylinder of an engine.

Taken in combination with a measurement of feed-water and coal used, the economy of the plant can be found.

Explanation of Important Points and Lines of Diagram.

Figure 111 shows the various important points and lines of an indicator diagram, the designations being as follows:

AE is the clearance line.

AN represents the amount of clearance.

CD is the atmospheric line. It is drawn by the pencil of the indicator, when its connections with the cylinder are closed and both sides of the piston are open to the atmosphere. It represents zero pressure of the steam gauge.

EF is the vacuum line.

HJ is the steam admission line.

J indicates the point of highest pressure.

JK is the steam line.

K designates the point of cut-off.

KL is the expansion line, showing decrease of steam pressure behind moving piston.

L indicates the point of release, i. e., the point where exhaust valve opens.

LM is the exhaust line, and shows loss of pressure where exhaust valve has been opened.

MG is the back pressure line.

G indicates the point of exhaust closure.

GH is the compression line, and shows rise in pressure due to compression of steam remaining in cylinder after exhaust valve has been closed.

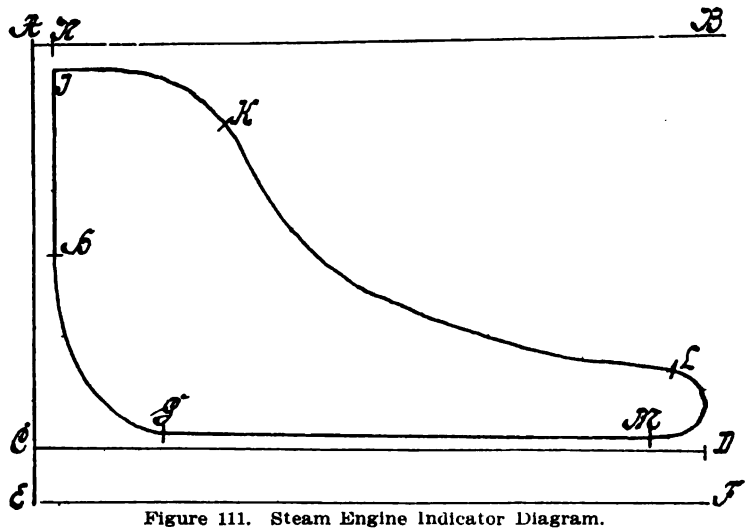


Figure 111. Steam Engine Indicator Diagram.

Theoretical Curve.

The expansion curve of the indicator diagram should coincide closely with the theoretical or *isothermal* curve, i. e., the curve of equal temperature. This curve is a common hyperbola, and must be laid off on the diagram, if the latter is to be examined regarding leakage of piston and valves.

There are different ways of constructing the isothermal curve. The following geometrical method for locating the points of the hyperbola is perhaps the one most generally used. Figure 112 makes this construction easily understood.

The *point of cut-off* G should first be located. At times this is difficult, but it may be located very nearly by simply noting the point where the curvature changes from convex to concave.

To draw the *vacuum line* the line EF is drawn at a distance corresponding to 14.7 pounds, below the atmospheric line. That is, if a 50-spring were used, the distance would be $\frac{14.7}{50} = 0.294$ = approximately $\frac{0.294 \times 32}{32}$ or 9/32 inch. The *boiler pressure line* AB is drawn in a similar way. The *clearance line* EA is

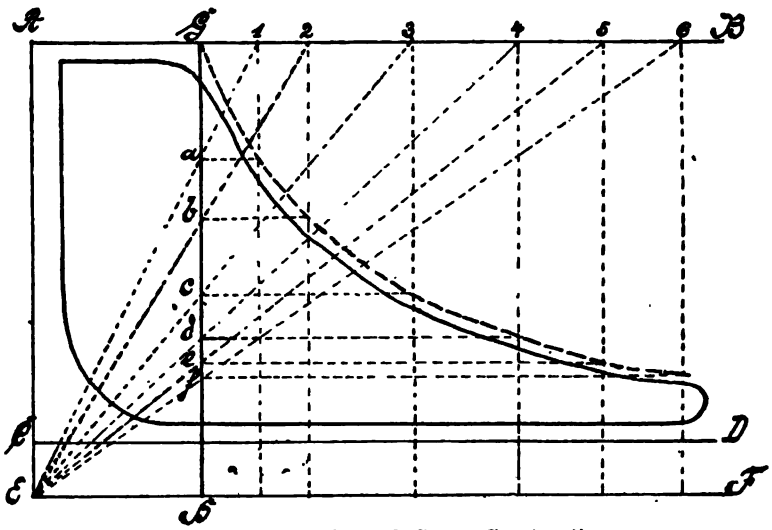


Figure 112. Isothermal Curve Construction.

drawn at a distance from the end of the diagram, corresponding to the per cent of clearance. For instance, if the clearance is 5%, and the diagram is 3¾ inches long, the clearance line must be drawn $0.05 \times 3\frac{3}{4} = 0.1875$, or approximately 3/16 inch from the end.

Starting from the point of cut-off, the pressure line is divided into any desired number of parts, which need not be of equal length. In Figure 112, the points of division are marked from 1 to 6. These points are connected with point E, and ver-

tical lines are drawn from them. The lines connecting the points with E intersect the line GH at a, b, c, d, e and f. From the latter points horizontal lines are then drawn, the horizontal a meeting the vertical line 1, the horizontal b meeting the vertical line 2, etc. The points of intersection of these horizontal and vertical lines are points of the hyperbola, and, connected, form the isothermal or theoretical expansion curve. In the illustration the theoretical curve is the heavily dotted line.

Determination of Mean Effective Pressure.

In order to find the mean effective pressure (M. E. P.) from an indicator diagram, the atmospheric line is divided into ten equal parts, as shown in Figure 113, or into twenty parts, if the diagram is very irregular. Half way between each two succeed-

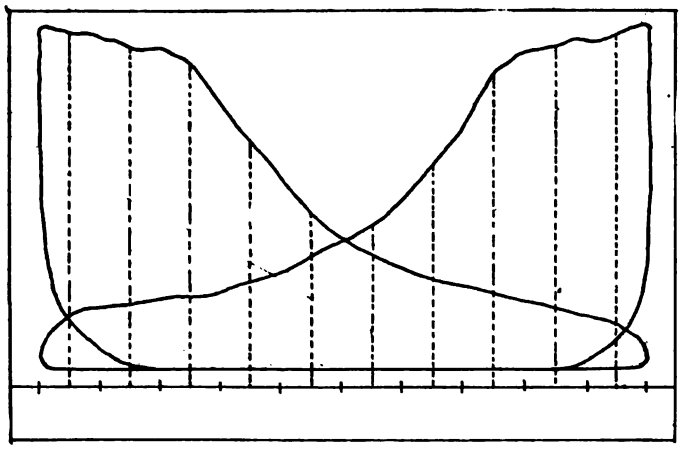


Figure 113. Mean Effective Pressure Lines.

ing points a vertical line is drawn, which is made to reach the height of the curve.

Each vertical line is measured, taking as the length the distance between the lines of the diagram, and not the distance from the atmospheric line. The lengths of the ten lines are added and divided by 10 (or by 20, if twenty divisions have been made). This gives the average length, which should be determined on the diagrams from each end of the cylinder. Adding the two values thus obtained, dividing by 2, and multiplying by the scale of the spring used, gives the mean effective pressure.

The work of determining the M. E. P. is greatly simplified by the use of a *planimeter*. This is an instrument designed to measure the areas of irregular figures. It is operated by moving a tracer, with which it is fitted, over the lines of the diagram or figure to be measured, and records the area upon a graduated wheel and vernier scale.

Determination of Theoretical Steam Consumption.

In order to find approximately from an indicator diagram the amount of steam consumed per horse-power per hour, the following methods may be employed:

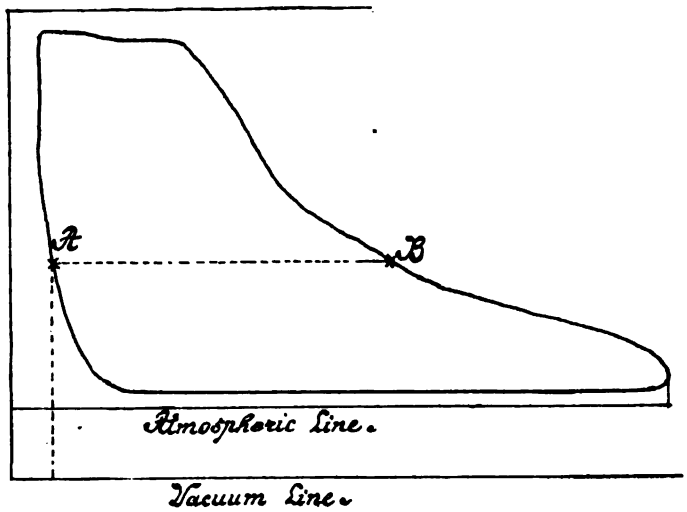


Figure 114. Determination of Steam Consumption.

Method 1.—Select two points on the diagram, Figure 114, equally distant from the vacuum line, one on the expansion line before release, point B, and one on the compression line, point A. Measure in inches the distance of points A and B from the vacuum line and multiply by the scale of the spring used, which gives the pressure in pounds per square inch at these points. From the steam table, page 252, find the weight of one cubic foot of steam at the pressure at A (or B), multiply this weight by the distance between A and B, multiply again by the constant 13,750 and then divide by the product of the mean effective pressure and the length of diagram.

Expressed in a formula, the above rule reads:

$$S = \frac{w \times d \times 13{,}750}{P \times L}$$

S = pounds of steam per indicated horse-power per hour.
w = weight of one cubic foot of steam in pounds at pressure at point A.
d = distance of point A from point B in inches.
P = mean effective pressure in pounds per square inch.
L = length of diagram in inches.

Example.—The distance between points A and B in the diagram, Figure 114, is 3½ inches, and their distance from the vacuum line is ¾ inch. The length of the diagram is 4 inches and the M. E. P. is 45 pounds. A 50-spring was used. How many pounds of steam were theoretically consumed per I. H. P. per hour?

Solution.—The absolute pressure at A (or B) is 0.75×50, or 37.5 pounds. From the steam table we find the weight of one cubic foot of steam at this pressure to be 0.0897 pound. Consequently the steam consumption is:

$$\frac{0.0897 \times 3.5 \times 13{,}750}{45 \times 4} = 24.$$

There are theoretically 24 pounds of dry steam used per I. H. P. per hour.

Method 2.—Another rule as given by Carpenter, which is more accurate, is the following:

Mark any desired point on the expansion line. To the percentage of the entire stroke, which has been completed by the piston at this point, add the percentage of clearance. Multiply this result by the weight of a cubic foot of steam of the pressure at the point in question. Subtract from this the product of the percentage of clearance and the weight of a cubic foot of steam at a pressure equal to that at the end of compression.

For simple engines multiply this result by 137.5, divided by the mean effective pressure. In the case of compound or triple expansion engines, divide by the equivalent mean effective pressure, on the assumption that all the work is done in one cylinder. Expressed in a formula, the above reads:

$$\text{Pounds of steam} = \frac{137.5}{p} \times [(b + c) \times w - c \times w_1].$$

p = mean effective pressure, in pounds per square inch.

b = percentage of stroke at point marked.

c = percentage of clearance.

w = weight of one cubic foot of steam in pounds at point in question.

w_1 = weight of one cubic foot of steam in pounds at end of compression.

For various reasons the calculated steam consumption is always less than the *actual* steam consumption. These reasons are, the presence of water due to cylinder condensation and not accounted for by the indicator, leakage through ports, radiation, etc. In a simple engine these losses usually amount to from 25% to 40%.

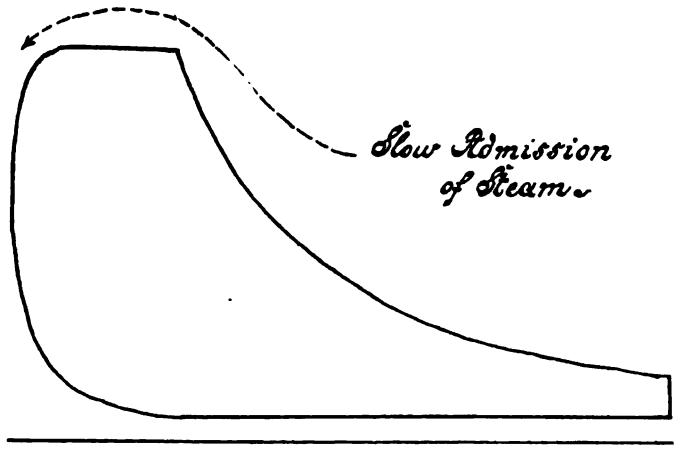


Figure 115.

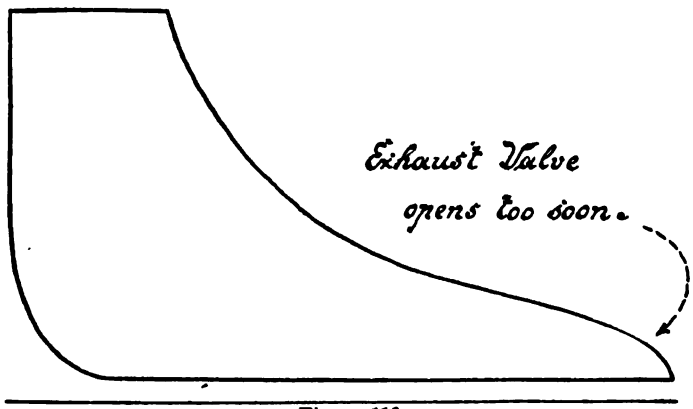


Figure 116.

Criticism of Indicator Cards.

Figure 115 shows a *slow admission of the live steam,* because the piston has traveled part of the stroke before sufficient steam opening was had to prevent throttling of the steam. In this case, either the lead of the eccentric or the lead of the live steam valve must be increased.

Figure 116 shows that the *exhaust valve opened too soon,* reducing the mean pressure and consequently the efficiency of the engine. The exhaust should not escape before the end of the stroke, but the round end proves that the exhaust valve opened too soon, allowing steam, which could have done work, to escape. In this case, the lead of the exhaust valve must be reduced.

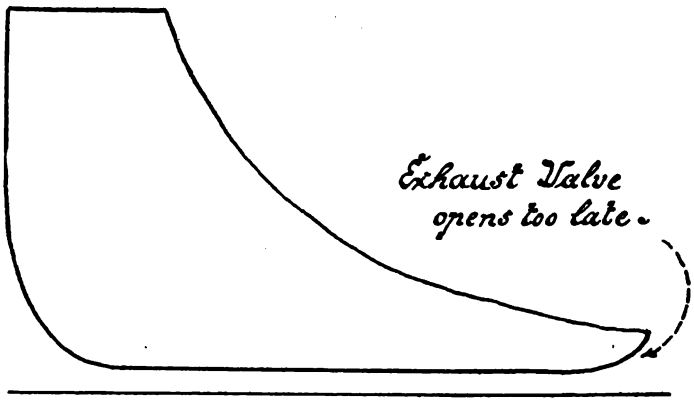


Figure 117.

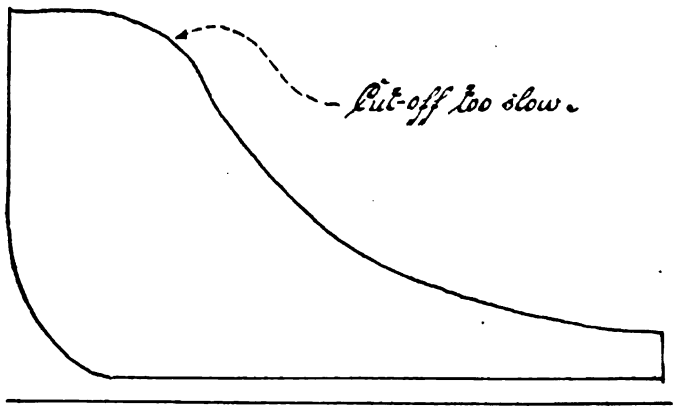


Figure 118.

In Figure 117 the *exhaust valve opens too slowly,* and the efficiency is reduced by compressing the exhaust, or better stated, throttling it. The pressure should drop in a straight line at the end of the stroke. In this case the lead of the exhaust valve must be increased.

Figure 118 shows that the *cut-off is too slow,* throttling the steam for a time instead of closing the live-steam valve quickly. This causes loss of efficiency, because the steam is not used altogether expansively. Either the stuffing boxes of the valve stems are too tight, or the valve has no oil, or the vacuum under the

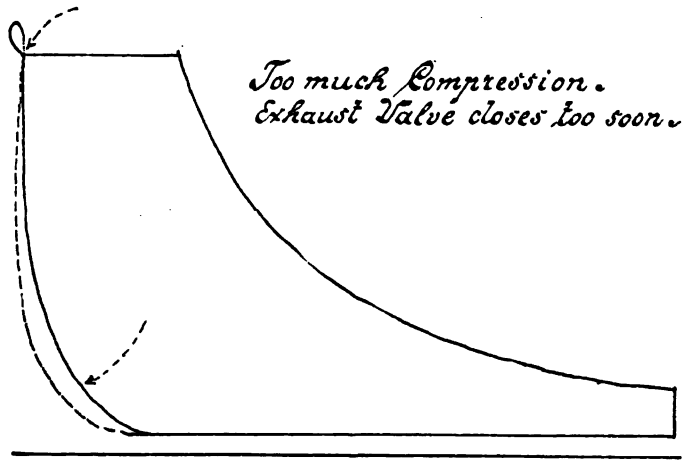


Figure 119.

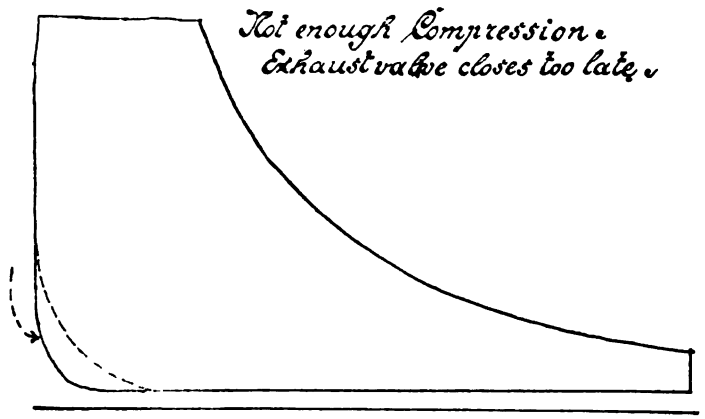


Figure 120.

dash-pot is not good. Should it be found that the first two causes do not apply, the last one must be looked into. It may also be that the dash-pot has too much cushion.

When the *exhaust valve closes too soon*, the compression may become higher than the boiler pressure. This is indicated by the characteristic loop in Figure 119. In Figure 120, the *exhaust valve closes too late.* Engines set this way are generally noisy, as the reciprocating parts do not receive sufficient cushion when reciprocation takes place.

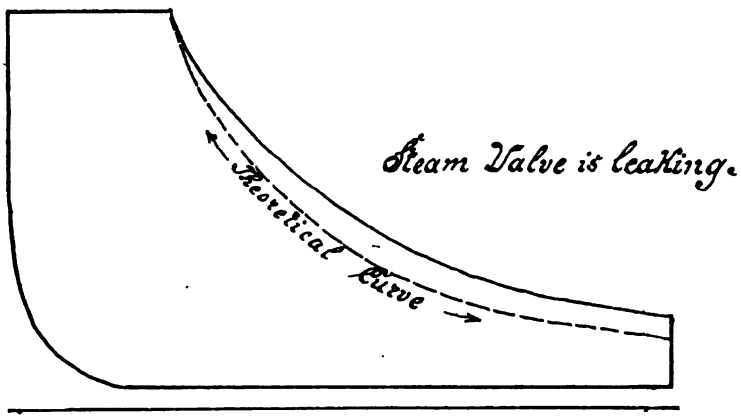


Figure 121.

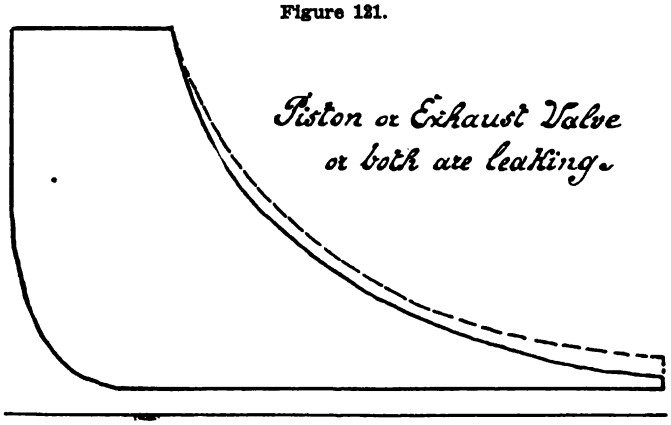


Figure 122.

In Figure 121, the *actual line lies above the theoretical line,* which indicates that there must have been steam added after the

live steam valve had been closed by the cut-off, and therefore, the live steam valve must leak and should be attended to.

Figure 122 shows the *actual expansion line below the theoretical line*. This can have been caused only by the steam leaking from the cylinder, or past the piston. Therefore the piston rings must be out of order or the exhaust valves must leak.

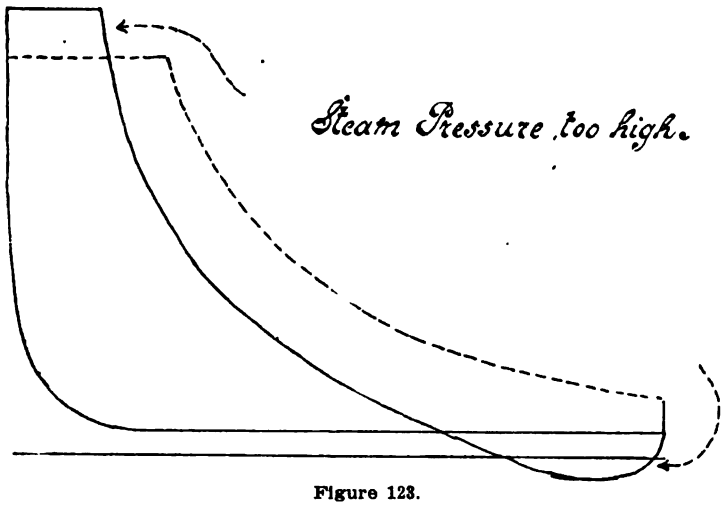


Figure 123.

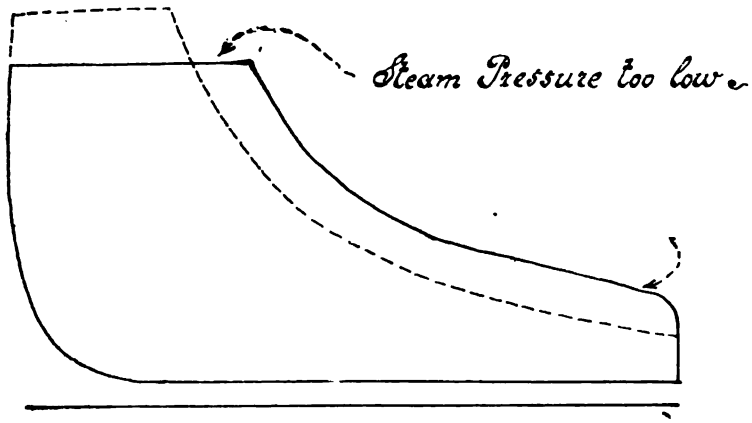


Figure 124.

In Figure 123, the *steam pressure is too high,* the cut-off is too early, and as there is more than the usual room for expansion

in the cylinder, the *steam expands below the atmospheric pressure.* This expansion, however, is lost, on account of atmospheric air rushing in when the exhaust valve is opened, thereby destroying much of the work previously gained by expansion.

In Figure 124, the *steam pressure is too low.* Consequently, the steam has no opportunity to expand sufficiently, and when the exhaust valve opens there is a considerable amount of steam pressure lost, which would be able to do useful work.

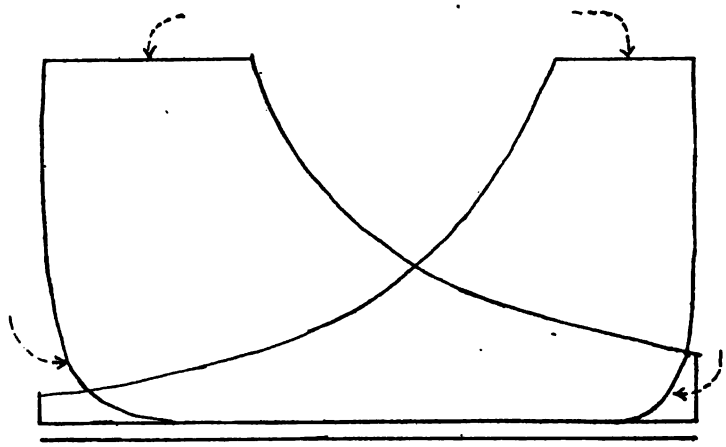


Figure 125. Unequal Cut-off and Unequal Compression.

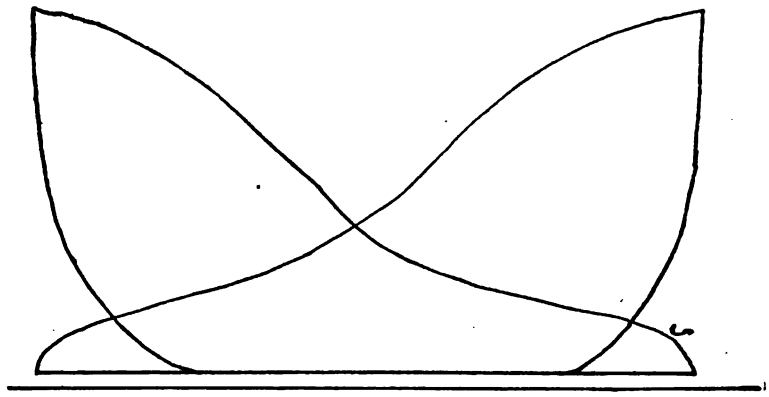


Figure 126. Card of High-Speed Engine.

Figure 125 shows *unequal cut-off and unequal cushion.* In the case of a Corliss engine, the cut-off should be regulated by

lengthening or shortening the governor rods, of course, after ascertaining which rod needs it. This can be done by raising the governor balls to the height the engineer knows they generally commence to act upon the cut-off. A piece of iron of the required height is generally kept for this purpose. It will thus be found which cut-off rod requires changing, which of course, is the one not cutting off under this position of the balls. The piece of iron can be placed under the ring sliding on the governor stem.

Figure 126 represents an actual diagram from a *high-speed engine* making 300 revolutions per minute with a piston speed of 600 feet per minute.

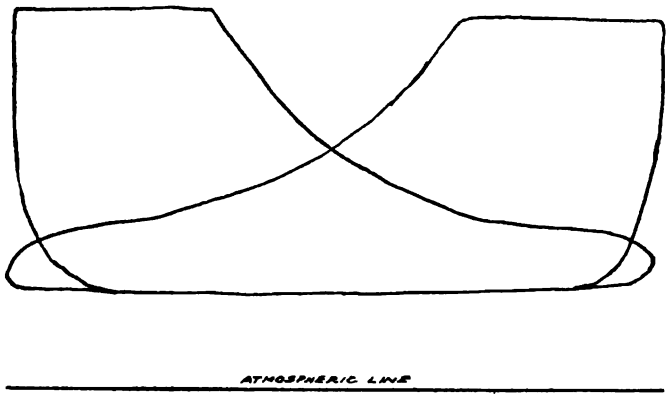


Figure 127. Card of High-Pressure Cylinder.

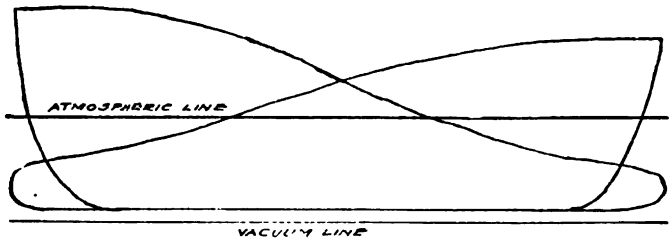


Figure 128. Card of Low-Pressure Cylinder.

Figures 127 and 128 give the diagrams of the *high-pressure* and of the *low-pressure cylinder* of an 18″ × 34″ × 42″ Hamilton Corliss compound condensing engine, making 102 revolutions per minute. It will be seen that the exhaust line of the low-pressure cylinder almost reaches the vacuum line. The diagrams may be considered good ones.

Steam Turbines

The main *advantages* of the steam turbine as compared to the reciprocating engine are : (1) small floor space and height, and inexpensive foundation; (2) simplicity; (3) no oil in the condensed exhaust steam; (4) no vibrations; (5) uniform rotary speed without use of fly-wheel; (6) capacity for considerable overload; (7) high efficiency for a large range of load.

The *disadvantage* of steam turbines in many cases, for instance for pumping stations and compressors, is the high rotary speed at which they have to be run.

In regard to *steam economy,* there is scarcely any difference between a high-grade piston engine and a first-class steam turbine in sizes up to 2,500 horse-power; for sizes larger than this, turbines are more economical.

There are two main classes of steam turbines: the *impulse* type and the *reaction* type. The impulse turbines may be either *single-stage* or *multi-stage,* while the reaction turbines are always multi-stage.

Impulse Turbines.

Single-Stage Impulse Turbine.—Figure 129 shows a section through a steam turbine of this type. The wheel W and the shaft are the only moving parts in the construction. D is the steam chest; from here one or more nozzles direct the steam against the blades of the wheel. After performing its work, the steam goes to the space F as exhaust steam and from there to a condenser or to the atmosphere. Figure 130 shows on a larger scale a section of a nozzle and part of the steam chest D. Figure 131 shows the shape of the blades, and how they are fixed to the wheel rim.

The action of the steam in producing turbine motion can now be followed. In the steam chest D (Figure 130), we have

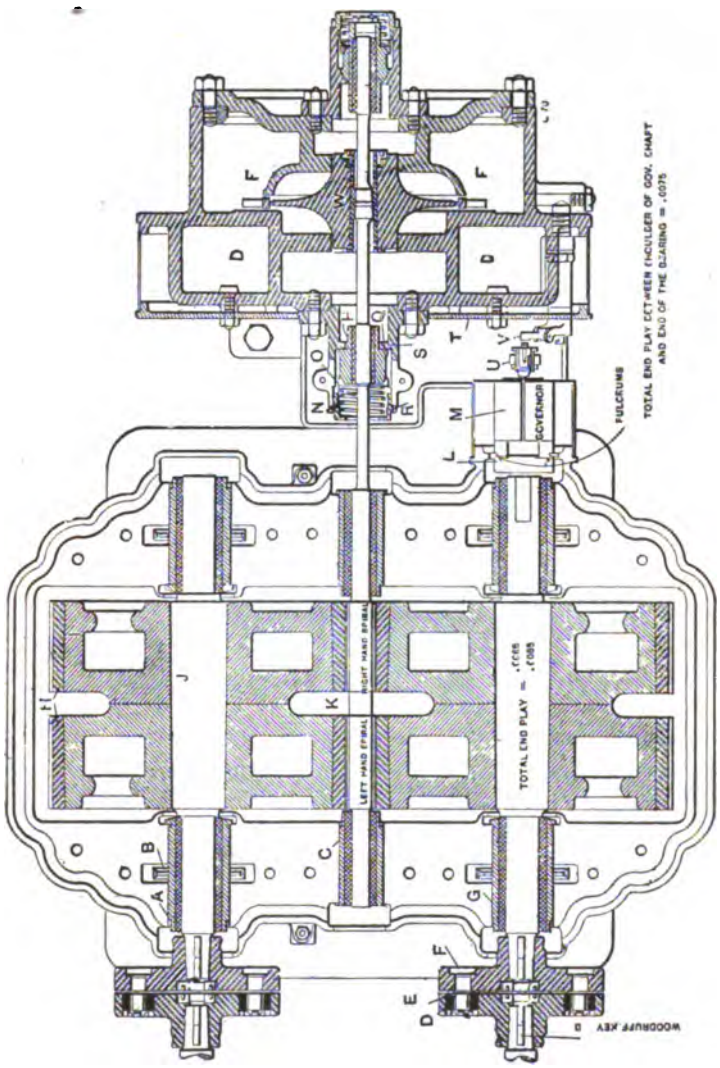


Figure 129. Single-Stage Impulse Turbine. (De Laval.)

steam of boiler pressure; at the other end of the *nozzle* B we have atmospheric or condenser pressure. Steam will therefore be forced through the nozzle at great velocity, the pressure decreasing during the passage. As the pressure decreases, the steam expands, therefore the nozzles are always designed diverging. The *blades* pass by the nozzle with high velocity, but not as high as the velocity of the steam. Therefore the steam impinges against the blades, rushes further between them and comes out in the direction indicated by the arrows, having given off most of its velocity, i. e., its kinetic energy.

The wheel is a steel disc, the blades drop-forged steel and the casing of cast steel. Some manufacturers call the blades *vanes,* others *buckets.* The shaft is very light and flexible; it

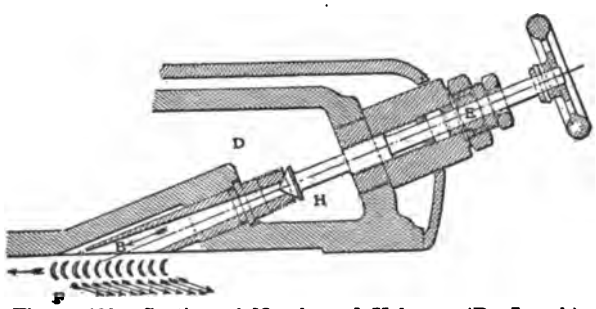


Figure 130. Section of Nozzle and Valve . (De Laval.)

has been found that this design gives much better satisfaction than a rigid shaft, as the wheel finds by itself the proper center of rotation. The shaft diameter, for instance, for a 300 horsepower turbine of this type is only 1-15/16 inches. The small steel helical pinion K (Figure 129) on the shaft transmits the motion through the large gears H to the two driving shafts, reducing the number of revolutions about 10 times. The outer ends of the blades are provided with flanges, which form a continuous outer rim, as shown in Figure 131.

The nozzles can be opened and closed by hand during running, and thereby the power output of the turbine is regulated. The number of nozzles is but one for the smallest sizes, up to fifteen for the largest sizes. The steam is used just as economically by a few nozzles, under low load, as by the full number of nozzles under full load. The smaller changes in load are taken

care of by a specially constructed shaft governor, which throttles the steam supply, when the speed begins to increase.

In this type of turbine the full drop in pressure, which is also the full drop in heat, is brought about in one stage. Therefore at the end of the nozzle we obtain a very high velocity of the steam, about 4,000 feet per second, depending of course on the initial pressure and condenser pressure. By calculation it can be proven that in order to get the maximum useful work out of the steam, the blades against which the steam impinges should have a velocity equal to half that of the steam. However, a velocity of the blades of $\frac{1}{2} \times 4{,}000 = 2{,}000$ feet per second has been found to be excessive and impracticable, the maximum in

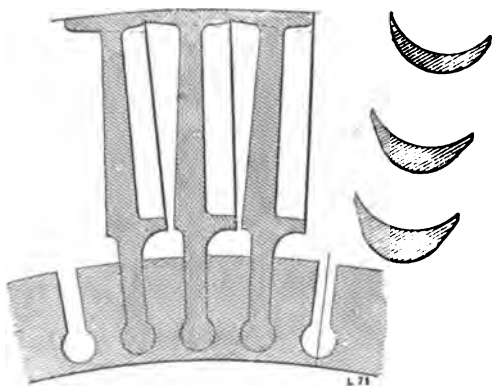


Figure 131. Blades. (De Laval.)

actual practice being about 1,300 feet per second. Therefore this type of turbine cannot show as high a degree of efficiency as the following types. It has other advantages, however, and has proven able to compete very successfully with the other types, in sizes up to 300 horse-power. For this size the wheel is 30 inches in diameter, and the speed is 10,000 revolutions per minute.

Multi-stage Impulse Turbines.—Figure 132 shows the principle in this construction, and the general arrangement of nozzles and blades for a two-stage turbine with two sets of moving blades in each stage. The nozzle diaphragm is a plate across the casing dividing it steam-tight into two separate parts. We therefore have three different pressures, the initial pressure in the steam chest, the condenser pressure after the second set of

nozzles, and a certain intermediate pressure between the two sets of nozzles. While in the single-stage turbine we have a pressure drop, for instance, from 160 pounds absolute to 2 pounds absolute through the nozzle, we will here have a drop, for instance, to 25 pounds absolute through the first set of nozzles and then from 25 pounds absolute to 2 pounds absolute through the other set of nozzles, and that means lower steam velocity at the nozzles.

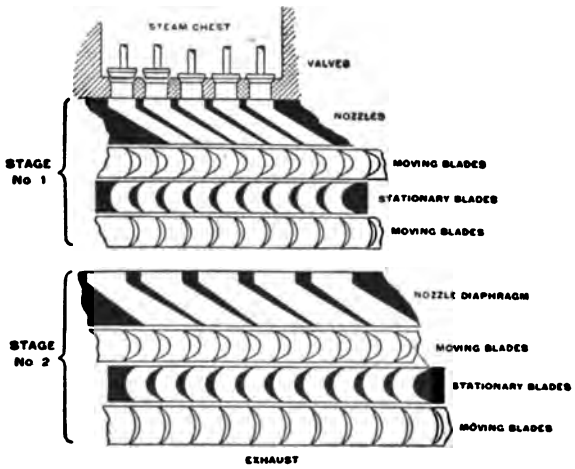


Figure 132. Section of Nozzle and Blades of Multi-Stage Impulse Turbine. (Curtis.)

The larger sizes of turbines of this type are divided into three, four, up to five stages, thus reducing still further the pressure drop through each set of nozzles. Each following set of nozzles must have a greater cross-sectional area than the preceding one, because the volume of the steam increases as the pressure decreases. Besides this division into stages, it will be seen from Figure 132 that inside of each stage the steam gives off its velocity to two rings of blades instead of one, as in the type mentioned above. These two rings of rotating blades are fixed on the same wheel, keyed to the shaft. Between the two sets of moving blades there is inserted a row of stationary guide blades, attached to the casing, which serve to alter the direction of the steam leaving the first set of moving blades, in such a manner that it impinges against the second set of moving blades

in the most efficient way. The larger sizes of turbines of this class have for each stage a wheel with three rings of rotating blades, and two rows of stationary blades inserted between them.

The purpose of division into stages is to get lower velocity of the steam leaving any of the nozzles. As stated above, the blade velocity should be about half of the velocity of the steam from the nozzle, if possible. In a two-stage turbine, under average conditions, the velocity of the steam leaving both sets of nozzles may be about 2,500 feet per second, and in a four-stage turbine the steam may leave each of the four sets of nozzles with a velocity of about 1,800 feet per second. Consequently the linear velocity of the blades should be 1,250 and 900 feet per second, respectively, a considerably lower speed than that required by the single-stage turbines for economical running. The purpose of letting the steam from the nozzles act on two or three sets of moving blades in each stage, is to have the velocity acquired by the steam, i. e., the kinetic energy, given off to the wheel more completely. Steam which still possesses a certain velocity, when it leaves the moving blades, represents a certain amount of kinetic energy, which has not been transformed into useful work.

Looking again at Figure 132, the total action of the steam through the two stages is as follows: The high pressure steam in the steam chest forces a continuous flow of steam through the first set of nozzles, leaving them with a velocity of say 2,500 feet a second. What intermediate steam pressure there is maintained in the first stage space depends on the exhaust pressure, on the number and size of the second set of nozzles and on the amount of steam admitted. The steam impinges against the first set of moving blades, giving off part of its velocity to these. Passing through the row of stationary blades, with the reduced velocity, it suffers no further loss in velocity or energy, and is only deflected so as to impinge against the second set of moving blades in the proper way. To these the steam gives off most of its remaining velocity. When the blades are correctly designed, the steam should barely have velocity enough left to get out from the moving blades.

Now again, this steam of a certain intermediate pressure forces a flow of steam through the second set of nozzles, as these are leading to the space, where condenser pressure prevails.

Leaving the nozzles the steam again has acquired a velocity of say 2,500 feet a second, and impinges against the first set of moving blades of the second stage. Part of the velocity of the steam is given off to these, and the steam, with reduced velocity, flows through the stationary guide blades, is properly deflected and impinges against the second set of moving blades of the second stage, leaving them with as little velocity left as practicable.

Formerly, the smaller sizes of the multi-stage steam turbines, up to 1,000 kilowatts, were usually of the horizontal type, while the sizes of 3,500 kilowatts and larger were of the vertical type. Recently, however, the vertical types are being supplanted almost entirely by the horizontal types, regardless of size. For use in electric power plants the arrangement is often as follows: The vertical steam turbine stands on the condenser as a base, and on top of it the electric generator is mounted. A 9,000 kilowatt steam turbine of this class runs economically at only 750 revolutions per minute, while the smallest sizes require 3,600 to 4,000 revolutions per minute.

Reaction Turbines.

Figure 133 shows a longitudinal section of a steam turbine of this class. It has no nozzles in the same sense of the word as used above. The action of the steam takes place in the three narrow, annular spaces L, M, and N between the stepped rotating drum, the *rotor,* and the corresponding stepped stationary casing the *stator.* The steam enters at A through a ring of stationary guide blades attached to the casing, part of which is shown in larger scale in Figure 134. Thereafter it passes through a ring of moving blades fixed to the rotor, then through another ring of stationary blades, then again through a ring of moving blades, etc., in this case through a total number of 62 elements or stages, each containing one ring of stationary and one ring of moving blades. Looking at the arrows in Figure 134, it may be said that there is, at any moment, between the rotor and the stator a number (in the medium sizes 200 to 300) of continuous, sinuous nozzles, through which the steam flows, and these nozzles increase in cross-sectional area from A towards the exhaust end. It is on account of this needed increase in steam passage area that rotor and stator are made stepped.

In the single-stage impulse turbine the whole drop in pressure and the whole production of velocity is done in one step. In the multi-stage impulse turbine there are from two to five distinct steps of pressure drop, producing velocity and giving off velocity to the moving blades. The principle in the design of the reaction turbine is that there shall be a gradual decrease in pressure, and therefore a gradual expansion and increase in velocity all the way through each of the continuous steam passages, the beginning of one of which is marked off by the arrows in Figure 134.

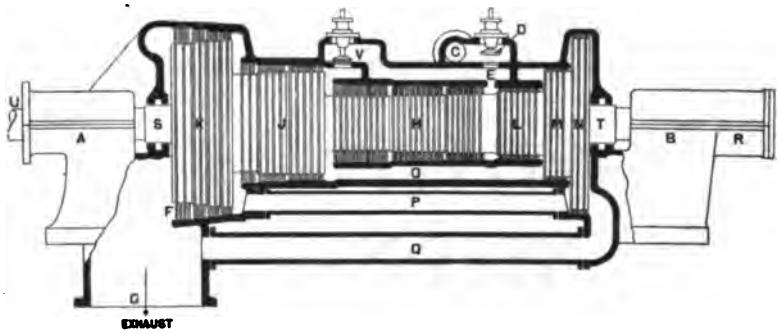


Figure 133. Reaction Turbine. (Parsons.)

The *action* is as follows: Between entrance side and exit side of the first ring of stationary blades there may be a pressure difference of 10 pounds or less; the steam expands and attains a velocity of say 400 feet per second (in this type it may be as low as 150). With this velocity the steam impinges against the ring of moving blades belonging to the same stage. Velocity, i. e., kinetic energy, is given off to the moving blades, but the passage is so designed that, while flowing through, the steam loses in pressure and therefore expands and receives velocity. The total result is that the steam leaves the first set of moving blades with a low velocity. On the exit side of the second set of stationary blades a still lower pressure prevails, the steam expands further, and a somewhat higher velocity is attained than in the first set of stationary blades. The steam impinges against the second set of moving blades, while passing through, velocity is both given off and received, and the steam flows away with low velocity.

The same series of actions is repeated through the whole number of stages, which may be as low as 10, and may be more than 100. On account of the low steam velocities in this turbine type, it can be run economically at a linear velocity of the blades of 300 feet per second, and even less. The speed varies from 3,600 to 750 R. P. M. The governor is often designed in such a way that while there is a practically continuous admission of steam at full load, the valve is momentarily closed at rapid intervals at lower loads. At considerable overload, an auxiliary valve is opened and admits high-pressure steam at a point where

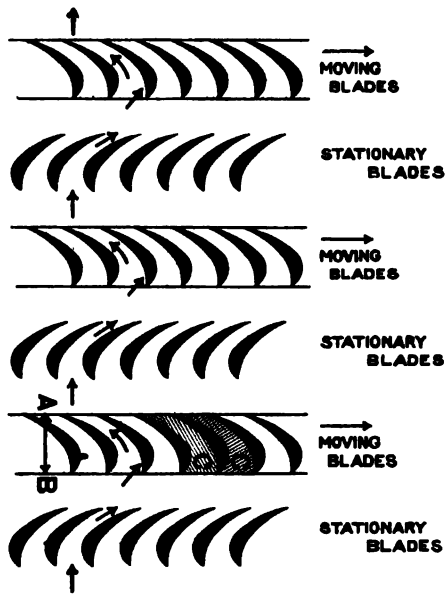


Figure 134. Section of Blades of Reaction Turbine. (Parsons.)

the area of the steam passages is greater than in the first ring of stationary blades, thus increasing the power of the turbine.

In the multi-stage impulse turbine there is no noticeable difference in steam pressure from entrance side to exit side either for moving blades or stationary blades. In the reaction turbine, however, there is, at any point, a certain difference in pressure between the space on one side of a ring and the space on the other side, and therefore in this type there is a tendency of the steam to leak over the ends of the blades, without doing any

work. For this reason it is of the greatest importance that the radial clearances, both near the casing and near the rotor, be made as small as possible, allowing for heat expansion. Another difficulty in the construction of this type is that the steam will have a tendency to press the rotor towards the exhaust end, as will be easily understood from Figure 133 and from the action of the steam already explained. It is for this reason, that the three *balance pistons,* often called dummies, are fixed on the rotor, rotating with it. They create exactly the same thrust in opposite direction.

The casing or stator is cast iron, while the rotor is a hollow cylinder of forged steel, in the smaller sizes a solid forging. The blades are of drawn brass, polished to make the friction as small as possible, but the first few rows are made of copper, in order to make it possible to use superheated steam. While in the turbine types with very high velocities there is a cutting action on the blades from the impinging steam, such does not take place in the turbines of this class.

Automatic Bleeder Turbine.

Many power plants, such as those in medium and large-sized breweries and various other industries, employ low pressure steam for heating purposes, and electrical current for power and lighting. The automatic bleeder turbine meets this double demand in a very economical way.

If the steam required for the heating processes is supplied to the mains from the boiler through a reducing valve, its expansive energy is wasted and its heating value is only slightly greater than that of ordinary exhaust steam. To eliminate this waste, the exhaust from non-condensing steam engines is often passed into the heating system and the expansive energy of the steam appears in the form of power. However, unless the demand for steam and power coincide, the gain is small, due to the low efficiency of the engine as a power producer. Another expedient is to supply the heating system from a receiver between the cylinders of a compound engine, but this arrangement is lacking in flexibility, since the receiver pressure varies with the power load on the engine; and in addition to this, the cylinder proportions of the engine are such as to give poor efficiency unless

a certain amount of steam is constantly passed to the engine out of the receiver. Such a close relation of the two demands scarcely ever exists in an industrial power plant, and as soon as either demand varies, operating difficulties are bound to occur due to over or under expansion, which will result in severe punishment to the engine.

The Westinghouse automatic bleeder turbine is practically free from the foregoing disadvantages, and will automatically supply a reasonable amount of low pressure steam to the heating system, and at the same time develop the required amount of electrical energy.

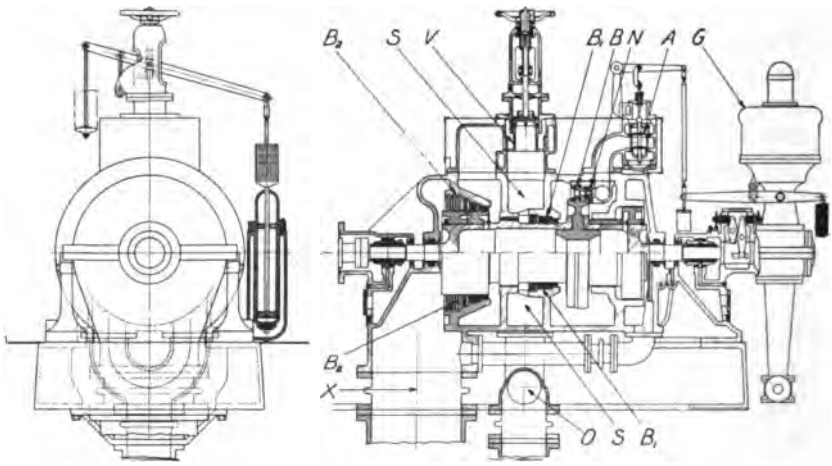


Figure 135; Westinghouse Bleeder Type Turbine.

The sectional view in Figure 135 shows the construction of the bleeder turbine. Steam from the boilers enters through the admission valve A, which is under control of the governor G. After suitable expansion in nozzles N, it passes through the high-pressure and intermediate blading B and B_1, where it produces power by taking the expansive energy from the steam. At the end of the blades B_1, the steam reaches an annular space S, from which it goes to the heating system through the outlet O. If the work done by the steam in passing through the blades is insufficient to carry the power load, the valve A will open wider, and more steam will pass into the turbine, raising the pressure at S. At V is a weight-loaded by-pass valve which opens when

this pressure is sufficiently great and allows the excess steam to pass into the low pressure blading B_2, where its expansion is completed before it passes to the condenser through the exhaust X. The loaded by-pass valve is provided, so that the pressure at which the steam passes to the low pressure blading, rather than to the heating system, may be varied as desired. For instance, a plant might carry five pounds gauge pressure on the inlet to the heating system during nine months of the year, then as the winter season made it necessary to heat the building, and the radiators were connected into the steam lines, it would be necessary to add weight to that already on the by-pass valve, so that no steam would pass to the low pressure blading until the pressure at S became that desired. It will be apparent that, except for changes of this kind in the weights, the operation of the turbine is entirely automatic, being exactly similar to that of any other turbine. If, for instance, it is desired to produce power without passing any steam to the heating system, this may be done by closing the valve in the line leading to the heating system, in which case all the steam entering the turbine passes through all the blading and thence into the condenser, or if desired, into the heating system through X rather than O.

Whatever the operating condition, the efficiency of the turbine as a power producer is very high, and is not affected by the bleeding out of steam.

A particularly desirable feature is that the steam so bled is free from contamination by oil, of which there is none inside the turbine, and the heating mains are, therefore, kept clean and free from obstruction by accumulation of grease.

Steam Turbine and Steam Engine Compared.

Referring again to advantages of the steam turbines stated in the beginning of this chapter, it will now be realized, that it is well warranted to mention simplicity as one of the advantages of the turbine. There are few moving parts, and no surfaces sliding on each other, except in the bearings. The question of lubrication is therefore also very simple, and this entails one of the other advantages, that no oil needs to come in contact with the steam. The condensed steam can therefore be re-used as feed-water without any trouble.

As regards economy in floor space, a reaction turbine or a multi-stage impulse turbine with accessories requires only about half of the space necessary for a vertical compound condensing engine of the same power. The combined outfit of electric generator and steam turbine has a weight which is, on an average, but one-eighth of the weight of the combined outfit of generator and reciprocating engine of the same power. Considering further, that on account of the absence of heavy parts moving up and down or back and forth, there is no vibration to speak of, it is evident that the foundations can be made much lighter. In many cases where reciprocating engines would require deep, massive foundations, the same space may, by use of steam turbines, be utilized as a basement for the condenser, and the turbine placed above the same on a few heavy beams, with concrete arches between.

Even with a heavy fly-wheel, there are some fluctuations in speed during each stroke of a reciprocating engine. Such is not the case in a steam turbine. Furthermore the governor has a more perfect control of the latter. Tests of several of the better makes show a variation in speed of 2 per cent from no load to full load, 3 per cent from no load to 100 per cent overload, and a still closer regulation can be obtained, if necessary. The best designed reciprocating engines have a variation of 2 per cent from no load to full load, and 4 per cent from no load to their highest overload. For the reaction turbine an arrangement for running at high overload was mentioned above. In fact, for any of the turbine types it is possible to make provisions to have it take care of an overload up to 100 per cent, without much loss in steam economy, while 50 per cent overload is the limit for reciprocating engines.

The first cost of steam turbines is about 10 per cent lower than the first cost of high-grade compound engines of the same power. As to the steam economy, there is, as mentioned in the beginning of this chapter, hardly any difference in sizes up to 2,500 horse-power, while above this size the tests show a difference in favor of the turbines. At rising back pressures, however, the steam consumption goes up quicker for the turbine than for the reciprocating engine. On the other hand superheating of the steam can be carried further in the turbine, and

both a very high vacuum (27—29 inches) and a high degree of superheating can be utilized more fully in the turbine than in the reciprocating engine. As indicator diagrams cannot be taken from steam turbines, the steam consumption cannot be given per indicated horse-power hour, but must be given per brake horse-power hour or, as is most generally done, per kilowatt hour (= 1.34 brake horse-power hour).

Internal Combustion Engines

Introductory.

The following data is designed to be of assistance to refrigerating engineers and is not intended to give more than a brief resumé of the subject. The data given is accurate as far as can be determined, and with the general working knowledge to be gathered from the following information the engineer will be in a position to obtain specific data relative to details of design, etc., from the numerous more complete treatises which are available.

General.

The internal combustion engine resembles the steam engine in that both are means of turning heat into work, the former being usually the more efficient as a prime mover, when it can be operated continuously at its rated load; it is not, however, considered as suitable as the steam engine for heavy overloads, and does not maintain its high efficiency on less than half load. Both types of engines utilize the force of expansion of a fluid, caused by heat, the internal combustion engine being self-contained in this respect, requiring no boiler. It is to be noted, however, that the gas engine is heavier than a steam engine of equal capacity, and requires a heavier crank shaft.

Types.

Internal combustion motors are divided into two types, 4-stroke cycle and 2-stroke cycle, commonly, but not correctly, called "4-cycle" and "2-cycle."

Engines of the 4-stroke cycle type have one explosion, and therefore give one power impulse every second revolution of the crank shaft, for each cylinder. The cycle of operations is as follows: 1) Suction or induction; the vacuum caused by the outstroke

of the piston drawing in the charge through the intake valve in the proper proportions. 2) Compression; the following instroke compressing the charge into the clearance space. 3) Explosion; the sudden expansion of the compressed mixture following ignition causing a rapid rise of pressure and driving the piston outward. 4) Exhaust or expulsion; the expanded gases being discharged through the exhaust valve by the returning piston. In some engines an auxiliary exhaust port is provided. This does not have a valve, but is located in the cylinder wall and is uncovered by the piston when near the end of the outstroke. This arrangement reduces the work of the exhaust valve.

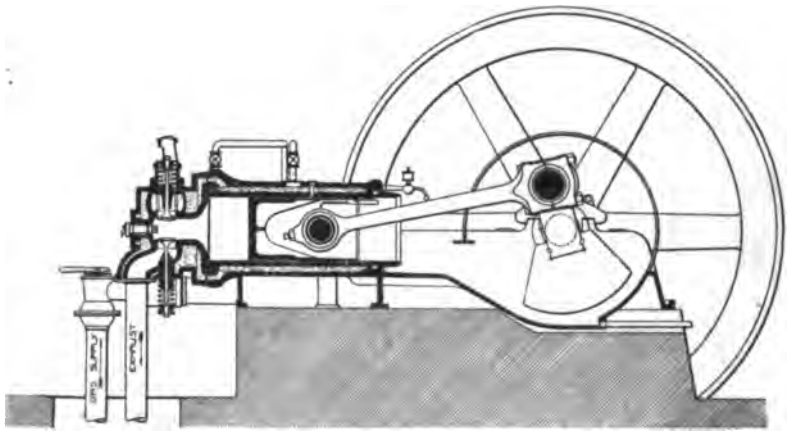


Figure 136. Four-Stroke Cycle Gas Engine.

Engines of the 2-stroke cycle type give one power stroke every revolution of the crank shaft for each cylinder. The 2-stroke cycle is as follows: 1) Compression; the instroke of the piston compressing the charge into the clearance space. 2) Explosion and expansion; the outstroke is the power impulse and opens the exhaust port near the end of the outstroke, and later the inlet port. As may be surmised from this description, a 2-stroke cycle engine requires auxiliary apparatus to prepare and inject the charge. This is usually done by auxiliary pistons working in separate chambers, or by enclosing the crank case and utilizing the slight suction and compression caused by the outer end of the piston. Some 2-stroke cycle engines are arranged so

that only air is admitted to the cylinder until after the exhaust port has been closed, thus assisting in scavenging or cleansing the cylinder of the exhaust gases and avoiding waste.

Internal combustion engines are usually provided with heavy fly-wheels, and the energy stored in these carries the load past the dead centers and facilitates speed regulation. The cylinders may be horizontal or vertical, and multi-cylinders are frequently

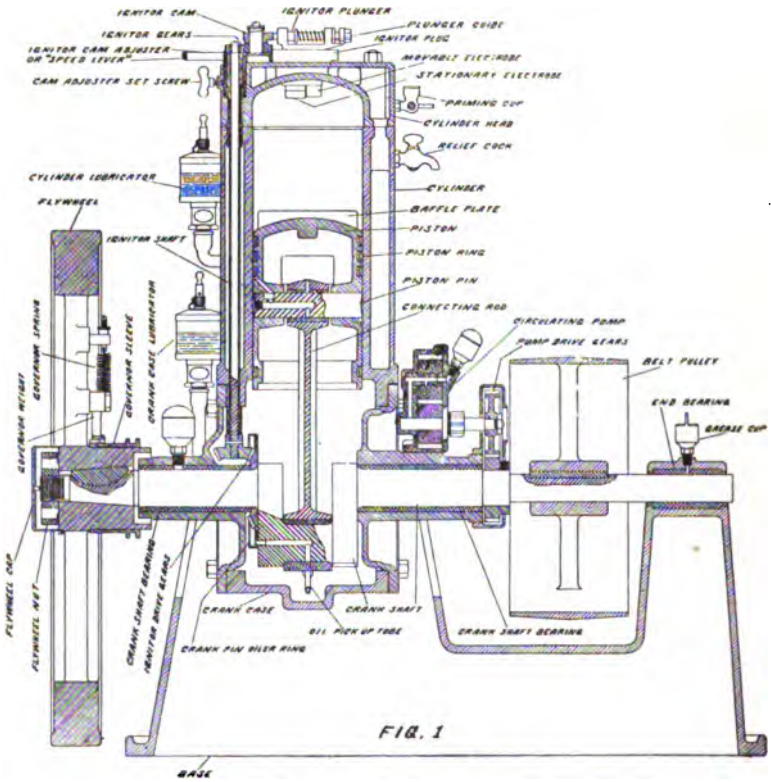


Figure 137a. Two-Stroke Cycle Engine.

used, especially for large units. Double-acting engines are also used, and these utilize both sides of the piston for receiving power impulses. This latter type introduces the complications of stuffing boxes and packing, subjected to high temperatures and pressures.

The 4-stroke cycle engines are generally conceded to be more efficient than those of the 2-stroke cycle type, having the advan-

tages of more complete combustion and lower internal temperatures, although they give fewer impulses and have larger cylinder dimensions per horse-power. Two-stroke cycle engines are, however, lighter per horse-power and give more impulses. Further-

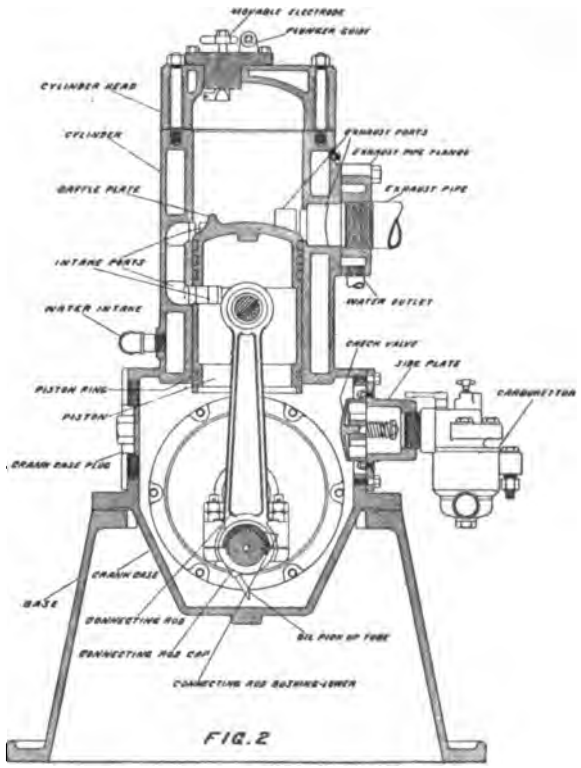


Figure 137b. Two-Stroke Cycle Engine.

more, they eliminate a large amount of valve mechanism, although they run hotter than 4-stroke cycle engines, and the increased speed gives greater wear on the bearings.

Valves.

Cylinder valves for internal combustion motors are usually of the poppet type, and are made of iron or steel, special steels being often used. Recent developments in design are bringing in rotary valves and also sleeve valves (as in the Knight motor).

These latter valves differ from poppet valves in that they do not leave their seats but rotate or slide until coincident openings form ports for the admission or discharge of the gases.

Simple 2-stroke cycle engines with crank-case compression eliminate all valves except one which governs the admission of air to the crank chamber. The crank-case compression is slight, being only about 4 pounds per square inch, and this valve is not subjected to heat. With double-acting 2-stroke cycle engines, or those with auxiliary compression chambers, control valves for the air are necessary, but again these are not subjected to as heavy duty as in 4-stroke cycle engines. Engines of the 2-stroke cycle type, having no cylinder valves, may be reversed by altering the timing of the ignition.

The valve gear of 4-stroke cycle engines is very important and is one of the main features in the reliable operation of the engine. The following discussion is limited to poppet valves. The valves are usually located in the cylinder head or in recesses generally cast integrally with the same, and may be vertical or horizontal, the former being preferred by many designers, who claim that this construction avoids uneven wearing of the seats and guides. Valve seats are usually beveled but are sometimes ground flat. In selecting an engine, care should be taken to see that the valves are easily accessible for removal and regrinding, if necessary, as leaky valves reduce the compression and therefore the power.

Valves are kept seated by springs, and exhaust valves are always opened mechanically. In some engines the intake valves are opened by the suction caused by the receding piston, but in engines designed for the best efficiency, the intake valves are also mechanically operated.

Mechanically operated valves are actuated by valve rods or lifters which receive their motion from a cam shaft. As each valve in each cylinder is opened once for every second revolution of the crank shaft, the cam shaft is necessarily geared to one-half of the engine speed. To reverse a 4-stroke cycle engine it is necessary to reverse the setting of the valves and therefore to change the location of the cams; the ignition must also be altered.

Mixers and Carburetors.

Engines operating on gas require only a plain control valve for the gas and a device for mixing the gas with the proper proportion of air (usually 8 to 12 volumes of air to 1 of gas). Almost any gas that will burn can be used for a gas engine; however, those usually employed are: natural gas, illuminating gas, producer gas and blast-furnace gas. Unless the cost is low, illuminating gas is usually employed only for small units or for intermittent service.

Carburetors are used to gasify liquid fuels and mix them with air before they enter the engines. Engines with carburetors are gas engines in every respect, but the gas is made in the carburetor and mixing passage instead of at some remote point. Carburetors are usually comparatively simple in construction, and when kerosene, alcohol or heavier oils are vaporized, the carburetor is usually heated either by the circulating water or by the exhaust. With heavier fuels than gasoline the compression of the engine is usually increased to give proper combustion. Carburetors do not truly gasify all of the fuel, but atomize part of it. The result, however, is an intimate mixture with air, which is what is desired.

Silencers.

Mufflers or exhaust pots are used to silence the exhaust. Open-type mufflers are placed at the end of the line, outside of the building. Exhaust pots are placed near the engine. Provision must be made for draining condensation, and mufflers and exhaust pots should be designed to safely withstand an internal pressure of 400 pounds per square inch. The principle of a silencer is to allow the hot exhaust gases to expand and cool before reaching the atmosphere. The water from the jackets is sometimes run into the exhaust pot to assist in cooling. Exhaust lines must be carefully insulated where passing through combustible floors and partitions.

Oil Engines.

Engines of the Diesel, Hornsby-Akroyd, and allied types are now extensively used on heavy fuel oils. The usual fuel is a distillate of crude oil from which the gasoline, kerosene and

lighter products have been removed. These engines are of either the 4-stroke or 2-stroke cycle type, and usually start the compression stroke with only air in the cylinder. The fuel is injected at the proper time, either by compressed air or by a suitable pump. The ignition usually results from the heat of compression or of previous explosions and will be discussed in a later paragraph.

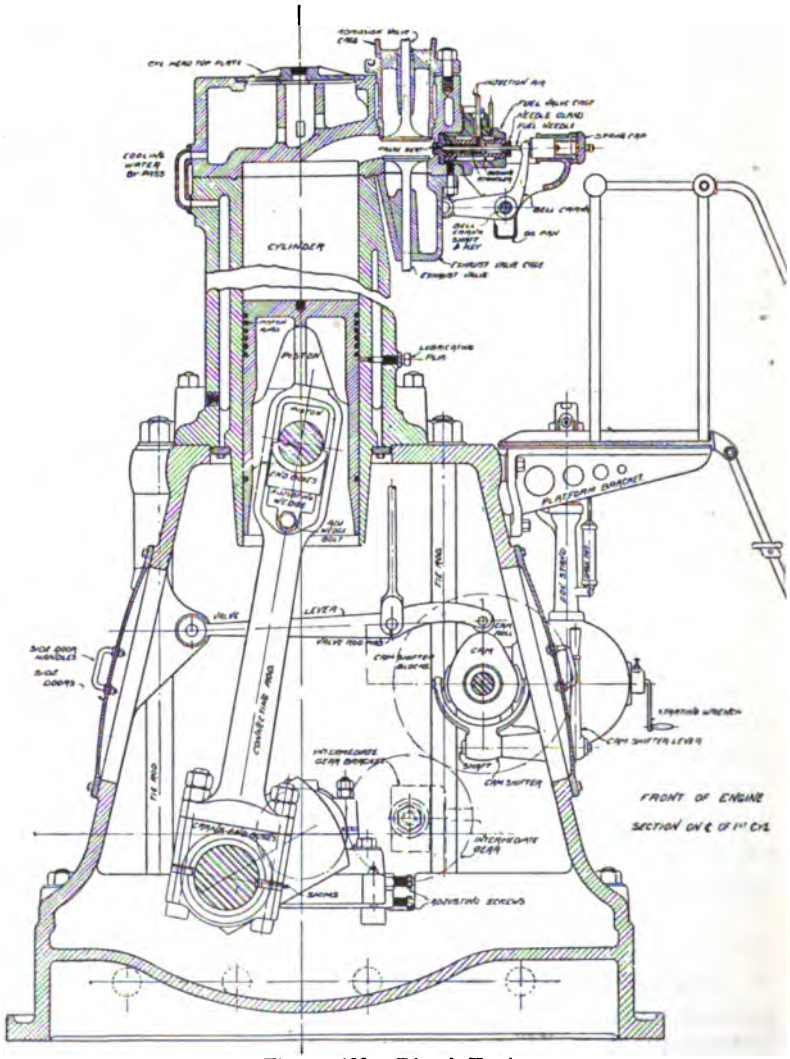


Figure 138. Diesel Engine.

Engines using fuel oils are usually heavier than gas or gasoline engines of the same power, as higher compression is necessary. They are cheaper to operate, however, the Diesel type being perhaps the most economical prime-mover in commercial use.

One of the inconveniences of using heavy oil engines is the difficulty of starting. Diesel engines may be started by compressed air operating in one cylinder, while the others receive the regular charge. Other oil engines must have the ignition chamber preheated by a torch (usually kerosene or gas).

Special spray nozzles for the fuel enter the cylinder in the clearance space, and, at the proper point in the stroke, a cam operates a pump which injects the proper charge into the cylinder in the form of a spray. This is either ignited by the heat caused by the high compression (500 pounds per square inch in the Diesel engine), or is directed against a projection or a vaporizing chamber which is kept hot by the heat of explosion and compression (in the Hornsby-Akroyd and similar types).

Ignition and Starting.

Almost all modern gas and gasoline engines employ electrical ignition. This may be either of the so-called "make-and-break" or the "jump-spark" type, the former being a low-tension arrangement, while the latter introduces high voltage.

A make-and-break igniter actually breaks a circuit inside of a cylinder. The spark caused at the moment the circuit is broken ignites the charge. There is one fixed insulated terminal and one movable terminal, and this introduces the necessity of a stuffing box where the oscillating member passes through the wall of the cylinder. The low-tension type is used on large units and also on a great many small engines; it is perhaps somewhat the more reliable type, but has a moving part passing through the wall.

A magnetic coil is usually introduced into the circuit to intensify the spark at the moment of interrupting the current, which may be supplied by batteries or by a low-tension generator or magneto.

The jump-spark or high-tension system is in general use on automobiles and also on high-speed engines. The engine con-

nections are made through spark plugs, and these are not subject to wear as are the parts of the make-and-break ignition. Ignition troubles are readily located and the plugs are easily removed. Recently a type of plug with an auxiliary spark gap and glass sights has come into general use, and thus defective ignition in any cylinder can be located without removing any connections. Current is supplied to the plugs either directly from a high-tension magneto or from a low-tension magneto through a coil which acts as a transformer and gives a voltage sufficiently high to jump the gap between the points of the plug.

Engines with electrical ignition may be started by turning the fly-wheels by hand in the direction of rotation, and compressed air is also often used for starting. In multi-cylinder engines, this is usually admitted to only one cylinder, a special arrangement of cams being used, and when the engine starts firing, the air is shut off, and the operating cams for the starting cylinder are shifted into place. Compressed air may also be used for starting single cylinder engines, sufficient momentum being stored in the fly-wheels before the air is cut off to carry the engine over a few working strokes.

Compression reliefs are usually supplied to facilitate starting by hand and priming is also often resorted to. With make-and-break ignition the cylinder can be primed, the fly-wheels turned backward to compress the charge, and the igniter tripped by hand to get the first explosion.

Engines using heavier oils, except kerosene, do not have electrical ignition. Some kerosene engines, however, are ignited electrically and may also be operated on gasoline in starting, being too cold to start on the heavier fuel. There are three usual types of ignition in use with oil engines, all of them being more or less related to each other.

In the Diesel type, the fuel is injected into the cylinder after the charge of air has been compressed. The high compression (about 500 pounds per square inch) raises the temperature of the air to such a point that the fuel ignites at once. The injection continues for about one-tenth of the power stroke, giving the effect of a longer impulse than would be obtained by a sudden explosion. No preheating is necessary with this type of engine, which is usually started with air pressure.

Another class of fuel-oil engines depends on the heat retained in a special chamber projecting outward from the cylinder head. With this class, the fuel is usually injected at the beginning of the compression stroke, after the exhaust is closed. As the compression increases, the mixture is forced into the ignition chamber, which is hot from previous explosions, and when the compression reaches the proper point, the mixture ignites from the heat of compression and the heat retained in the walls. The special ignition chamber of these engines must be pre-heated and this is usually accomplished by gas or kerosene blow torches. When the chamber is sufficiently hot, the fly-wheels are turned backward by hand against the compression, the oil pump being operated by hand, injecting into the cylinder a charge which is at once ignited, thus starting the engine.

The third class of fuel-oil engines is very similar to the second except that a hot spoon or ball projecting into the cylinder is used instead of the ignition chamber. With these engines, the fuel is injected later than with the previous class and is sprayed directly against the hot projection, the compression being high enough at the time of injection to effect ignition. These engines are started in a manner similar to that described for the second class of fuel-oil engines.

Governing.

Gas and gasoline engines are either governed on the so-called "hit-and-miss" principle or by some type of throttling governor. The former type is more frequently used on small engines. There are some variations in the design of the "hit-and-miss" governors, but they usually cut out the ignition and hold the exhaust valve open when the engine runs above the proper speed.

Throttling governors vary the quantity or quality of the mixture or both. When the quantity of the mixture is altered, the compression is of course varied, and is therefore reduced at light loads.

The power to operate the governing mechanism is usually derived from the centrifugal force generated by weights which are located on the fly-wheel or on a pedestal, as with steam engines.

Kerosene and oil engines may be governed by varying the fuel supply or on the "hit-and-miss" principle. These engines

may sometimes be governed by varying both the fuel and the air, but the general practice is to supply an excess of air.

The fuel supply is regulated by by-passing varying amounts of the oil supplied by the injector pump or by varying the stroke of the pump. When the "hit-and-miss" principle is used the exhaust valve is held open, thus destroying the compression.

In order to overcome the cyclical speed variation at each revolution without resorting to fly-wheels of excessive weight, it is usual to employ multi-cylinder engines, to obtain greater frequency of impulse. The variation due to alteration in the load must be taken care of by the governor. The most exacting work in the matter of speed regulation is the operation of alternating current generators in parallel, and this is being very successfully accomplished.

Cooling.

Owing to the very high temperature in the cylinders, all internal combustion engines must have some form of cooling system. The object of the cooling system is to keep the piston and the cylinder walls at such a temperature that they will not seize, and so that the oil will not become overheated. The cylinder must not become too cool or a loss of efficiency will result. The best operating temperature is about 180° Fahrenheit.

When a water-cooling system is used, the cylinder walls and heads and the valve chambers are water-jacketed, the jackets usually being cast integrally with the cylinders. On some large units the piston head is also cooled. The water is either run through the jackets and wasted or is returned to a cooling reservoir either by syphonic circulation or by a pump driven by the engine. In case of syphonic circulation the water level in the tank must be higher than the engine cylinder.

With air-cooled engines, ribs or vanes are cast integrally with the outside of the cylinder, and air is blown against these by a fan, the cylinder sometimes being jacketed with sheet metal to force the air against all the vanes or projections. Air-cooling is not used for large engines, but is of service where light weight is essential.

Great care must be exercised to prevent freezing of the water

in the jackets, as this is very liable to result in cracked cylinders. After stopping in cold weather, the jackets should always be drained. Some manufacturers recommend the use of oil instead of water for cooling. This has a higher boiling point and will not evaporate rapidly, and, furthermore, has a much lower freezing point than water.

The temperature of the cooling water is an indication of the condition of the engine. If the water is too cool, the engine is not being operated at its maximum efficiency. If the water is too hot, the engine should be examined at once. The circulation may be impeded, the lubrication may be insufficient or the oil unsuitable, or the mixture may be too rich.

When overheated, an odor of burning oil and hot metal may be detected and slight smoke may be seen. There will also usually be a knocking sound in the engine. Whenever this is the case, the engine should be stopped at once. The engine should be turned by hand after injecting kerosene in large amounts into the cylinder. This is to free the piston rings and prevent seizing of the parts. A simple test for overheating is to drop a little water on the cylinder head. If it vaporizes immediately, the engine is too hot and should be stopped for adjustment; if the water sizzles for a few moments, the case is not so bad, but whenever overheating is suspected, caution should be used and the lubrication, the circulation and the mixture inspected.

Lubrication.

Proper lubrication and suitable oil are essential to the efficient operation of an internal combustion engine. Sight or pressure feed or both may be used, an advantage of the latter being that the feed is automatically increased as the engine speeds up. The splash system is readily applied to vertical engines.

The most difficult part to lubricate properly is the cylinder, particularly in gasoline and kerosene engines. Owing to the high temperature and the high piston speed, it is essential to use carefully selected oil. Animal and vegetable oils should never be used in the cylinder. These oils may decompose at high temperature and the resultant acids affect the parts.

Good pure mineral oil should be used for the cylinder. High flash point and ignition point are essential. A very good cylinder

oil is one with a specific gravity of about 0.89 at 62° F., and a flash point of over 500° F. When the engine may be exposed to cold, attention must be paid to the temperature at which the oil solidifies. There are good oils on the market which remain liquid at quite low temperatures.

Although the temperature of explosion is well beyond the ignition point of the cylinder oil, it does not burn rapidly if the water circulation is good, as it is spread in a thin layer on the comparatively cool cylinder wall. If the cylinder is not properly lubricated, loss of compression or seizing of the rings and piston may result.

The bearings are lubricated by grease cups, ring oilers, sight-feed or forced-feed systems. The time spent in attending to lubrication is well repaid. Lubrication should have regular attention and should never be neglected. Too much oil should be avoided, as this tends to foul ignition, accumulates carbon in the cylinder, and causes smoke at the exhaust. This can be detected by dense white smoke, sometimes with a yellowish tinge, at the exhaust, and the smell of burned oil. Dark smoke indicates a large excess of fuel, a smaller excess being shown by white smoke and a characteristic pungent odor.

The makers furnish specific directions for lubricating the various parts and care must be taken to see that the instructions are properly followed. The oil tubes and passages must be clean so that the lubricant has free access to all of the parts. Care should be taken to see that the lubricators are closed when the engine is stopped. When starting, the grease cups should be turned up and the lubricators put in commission according to the manufacturer's directions.

Gas Producers.

Owing to modern development of the gas producer, gas engines may be operated independently of natural or other artificial gas supply. When contemplating installations of 75 H. P. and over, it will usually pay to look carefully into the adaptability of a producer.

Although a complete gas power plant would cost more than a steam plant of equal capacity it has a number of advantages.

A gas plant transforms 15% to 22% of the heat value of the fuel into work, as compared with 5% to 12% with steam plants. A small gas-power plant compares favorably in heat efficiency with very large steam plants. Gas plants are much cleaner, require less labor, and the stand-by losses are smaller. A wide range of fuel is available for a producer plant: coal, coke, sawdust, tan-bark, etc. It is necessary to carefully cleanse the gases

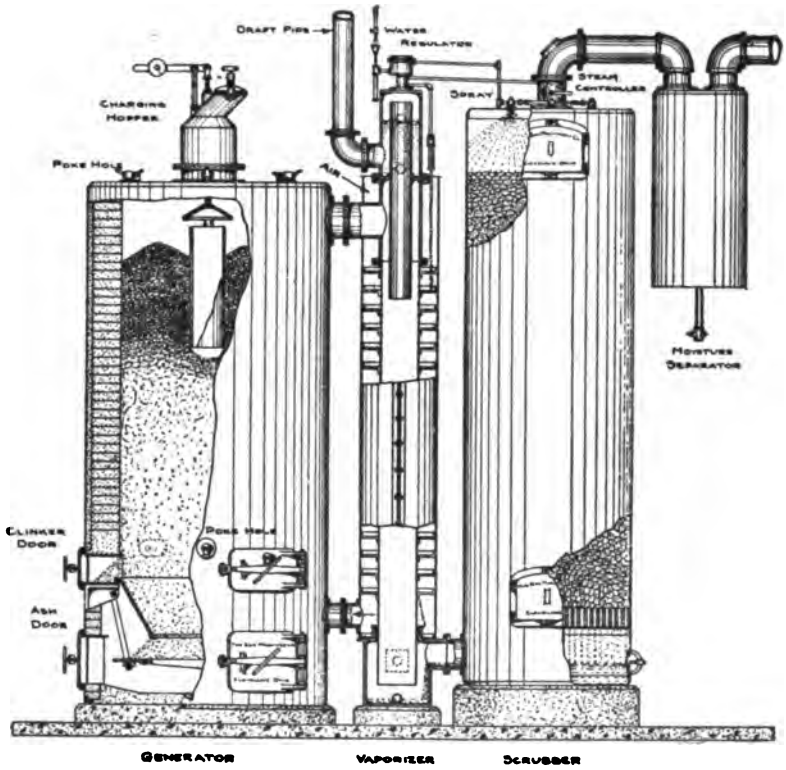


Figure 139. Suction Gas Producer.

when used for engines, and therefore some users favor the anthracite producer, as the entire plant is then comparatively simple. Some bituminous producers are, however, arranged so that the impurities are burned when the dirty gas passes through an incandescent fuel bed, and bituminous fuels are usually cheaper.

Producers are made in either suction or pressure types, or in combinations of both.

The action of a producer is to burn the fuel with a limited supply of oxygen and in some cases in the presence of water vapor, thus obtaining incomplete combustion, and making a gas containing a considerable amount of carbon monoxide. Anthracite coal, for example, gives a gas containing about the following:

Carbon dioxide (CO_2)	2.5%
Carbon monoxide (CO)	28.0%
Hydrogen (H)	9.0%
Marsh gas (CH_4)	1.0%
Nitrogen (N)	59.5%

Producer gas is usually a comparatively lean gas, averaging about 130 B. T. U. per cubic foot. It is to be noted that a complete gas plant will require about 15 gallons of water per brake horse-power hour. This is about three times the amount needed for a non-condensing steam plant, whereas a condensing plant uses about 60 gallons. If the water is re-cooled and not run to waste, both the steam and the gas plants can operate with a consumption of not over 2 gallons per actual horse-power hour.

A gas producer consists essentially of a generator, a wet scrubber, dry purifiers, and sometimes auxiliary apparatus such as economizers, gas tanks, etc. The generator is usually a metal cylinder lined with fire brick and fed from the top. The wet scrubber is a similar cylinder filled with coke, broken stone or lattice work, over which water is sprayed from the top, the gas entering at the bottom. The dirt and tar are washed out of the gas and run into the water seals. Dry scrubbers are tanks which usually contain wood shavings and iron oxide or other purifying materials. Economizers are sometimes used to remove some of the heat from the gas between the generator and the scrubber, and to utilize this heat to vaporize water for the generator and to preheat the incoming air for the generator.

Utilization of Waste Heat.

A practical problem to be considered when installing internal combustion engines for refrigerating machinery is the utilization of the cooling water and the exhaust for heating purposes. Of course, this heat is only available when the engine is in operation. Sometimes low-pressure boilers are installed for auxiliary

heating, and often, where considerable steam is necssary in a plant, only a part of the power is derived from internal combustion engines.

There are various types of apparatus on the market to utilize the waste heat from such engines, and the heat is either applied by water circulation, by circulation of air heated by hot pipes, or by steam. Exhaust heaters usually have the exhaust pass through an enlarged chamber containing water pipes leading to the heating system. At ½ load, only about 60% as much heat is available as at full load.

Operating Hints.

Safety.—Keep a box full of fine dry sand in the engine room when operating on liquid fuel. Keep a big scoop in the box and keep the box covered. One pint of gasoline will make 200 cubic feet of air explosive. If small amounts of hazardous fluids are necessary, keep them in approved safety cans. Incandescent lamps in vapor-proof globes are preferable for lighting engine rooms. Do not pour water on fires in petroleum oils. Water is, however, of assistance on alcohol fires. Be sure that all gas connections are tight and that the fuel valve on a gas engine is closed when the engine shuts down. Valves which automatically shut off the gas supply when the engine shuts down for any cause are desirable. Never test for leaks with a match, use soapy water.

Care.—Keep the engine clean. Do not allow dust and dirt to accumulate. Filter lubricating oil, and be sure gasoline is free from dirt and water. Filtering gasoline through chamois removes water.

Ventilation.—Engine rooms should be ventilated at both top and bottom. Gas is lighter than air, but gasoline vapor is heavier.

Cylinder and Piston.—Kerosene or turpentine are of service in removing carbon. Never scrape cylinders or pistons with hard tools. Use soft copper or brass.

Joints.—Metal-to-metal joints may be made tight with a little red lead. When gaskets are necessary, woven asbestos reenforced with fine wires and treated with graphite is excellent.

Valves.—Kerosene is of service on valve stems to keep them clean and free from carbon. If the springs become weak, they should be replaced. Unless valve seats are so badly injured as to require machining, they can be ground by hand. Special abrasives for grinding are available. Valves should be turned first one way and then the other, and should not be rotated all the way around when re-grinding.

Dismanteling.—When taking down an engine, note carefully the position of the marks on the gears, etc., so as to reassemble the apparatus properly timed.

Nuts.—Use spanners or socket wrenches where possible. Very tight nuts may often be loosened by heating, using an iron bar or pipe bent into the form of a spanner.

Cooling Systems.—About 5 or 6 gallons of water per horse-power hour should flow through the jackets. Anti-freezing solutions may be made by the use of salts, glycerin or alcohol, or combinations of these in the cooling water; alcohol, of course, being replenished as it evaporates. Hard waters may form incrustations on the jacket walls, throttling the flow and insulating the cylinder. Scale can be loosened by a dilute hydrochloric acid solution, allowed to stand several hours in the jacket. The jacket must be carefully washed out afterward. It is good practice to drain and wash out the water passages frequently.

Weak Compression.—If the compression is weak, the engine will not run efficiently. The causes are usually leaky valves, or leaks past the piston due to poor lubrication.

Overheated Bearings.—If very bad, the engine should be stopped and the bearing cooled. Accessible bearings should be felt by the hand from time to time, and extreme care should be used to see that all bearings have sufficient lubrication. If a bearing is too tight, insert a thin plate or shim between the parts so that the bolts can be kept tightly set up.

Wiring.—Keep the wiring clean and away from oil which may rot insulation. Wiring should be firmly held to prevent abrasion and loosening.

Ignition.—If too early, the engine may back-fire. The engine should be properly timed. After starting, the spark should be advanced and the fuel supply cut down to a point at which the

engine can properly carry its load. Avoid racing the engine, and avoid very sudden large changes in the load if possible. If much ignition adjustment is necessary, it would be well to use an indicator and have an expert inspection made.

Priming. — Be careful when priming directly in the cylinder, as gasoline cuts oil, and an excess may remove too much of the lubricant. A good place to prime is at the intake manifold, close to the cylinder.

Transmission of Power

In general, power is transmitted by means of shafting, but various devices are employed to transmit it from one shaft to another. Among these are belts, gear wheels, chains, ropes and friction pulleys. In order to be able to transmit a given amount of power, the material used must possess a certain strength, that is, it must be able to withstand the stresses acting on it, without deformation.

Strength of Materials.

Stress and Strain.—Much difference of opinion exists as to the proper use of these two terms. An external force acting on a body, tends to deform it, and is resisted by an internal force. The terms stress and strain are applied to either of these forces. In this book we will only consider the term stress, and use this as applying to both the external force applied, and the resisting internal force.

When a force acts on a body, there are five different stresses which may arise, namely, *tensile, compressive, transverse, torsional,* and *shearing* stresses. A *tensile stress* arises in a piece when it is subjected to pull (for instance, in a vertical rod carrying a weight); a *compressive stress,* when it is subjected to pressure (for instance, in a washer under a nut); a *transverse stress,* when it is subjected to bending (for instance, in a rod supported at the ends and carrying a load between the supports); a *torsional stress,* when it is subjected to twisting (for instance, in a fly-wheel shaft); and a *shearing stress,* when a force tends to slide part of the piece over the remaining part (for instance, in shearing of rivets).

Strength.—If a vertical bar of 2 square inches cross-sectional area carries a weight of 20,000 pounds, then there is in the bar a uniform tensile stress of 10,000 pounds per square

inch. If the weight is increased, the tensile stress increases in the same proportion, and if it is found that rupture occurs when the weight has been increased to 100,000 pounds, it is said that the material of this bar has a *tensile strength* (sometimes called *ultimate tensile strength*) of 50,000 pounds per square inch. Similarly, the compressive, transverse, torsional, and shearing strengths of a material are the respective stresses in pounds per square inch at which breakage occurs.

Elasticity.—Under small stresses a vertical rod, subjected to tensile stress, undergoes an elongation, proportional to the length of the rod and to the force applied, and inversely proportional to the cross-sectional area of the rod. When the load is removed, the body will, in most cases, return to its original form, due to elasticity. When the stress is sufficiently great the body does not resume its original form, and is said to have a *permanent set.* The point at which this set occurs is termed the *elastic limit.* More correctly, however, the elastic limit should be defined as the point at which elongation increases at a more rapid rate than the stress.

It was once the belief that stresses inside of the elastic limit would not under any circumstances cause permanent changes, either in the form or in the qualities of a piece of material. It has, however, been proven that if a piece is alternately subjected to a stress and relieved again, a great many times, a break is liable to occur, even if the stress has been kept within the elastic limit.

Coefficient of Elasticity.—By coefficient or modulus of elasticity is meant the ratio of the load per unit of section to the elongation per unit of length. For steel it is 30,000,000, and for wrought iron 25,000,000. The formula for elongation of a piece subjected to tensile stress, within the elastic limit, follows:

$$E = \frac{P \times l}{k \times C}$$

E = elongation in inches.
P = load in pounds applied on the rod.
l = length of rod in inches.
k = cross-sectional area of rod in square inches.
C = coefficient of elasticity.

Example.—A steel rod 10 feet long and 1 inch in diameter is subjected to a tensile stress of 5,000 pounds. What is the elongation?

$$Solution.—E = \frac{5{,}000 \times 10 \times 12}{0.7854 \times 1 \times 1 \times 30{,}000{,}000} = {}^{1}/_{39} \text{ inch.}$$

Factor of Safety.—By factor of safety is meant the ratio of the load that is just sufficient to cause rupture in a piece of material, to the maximum load ordinarily acting on it. If, for instance, a pulling force, which under average conditions reaches a maximum of 20,000 pounds, acts on a piston-rod, and if it is known that the rod would rupture the moment the force equaled 300,000 pounds, then the factor of safety for the rod is

$$\frac{300{,}000}{20{,}000} = 15.$$

Both for structural pieces in a building and for machinery parts, we speak about *live loads* and *dead loads*. By the former expression is meant a changeable load, causing changes in the stresses. The piston-rod mentioned above is an example of a piece under live load. There is, of course, much difference in the extremes to which the changes go, and as to how quickly they follow each other. The piston-rod is subjected to more severe service than most machinery parts, excepting parts that are exposed to shocks during operation, and there are many cases of live loads where it would not be necessary to use a factor of safety as high as 15. Pipe hangers carrying a water pipe that is always filled with water and not exposed to shocks or vibrations, are an example of dead load. Here a factor of safety of 3 is used, and authorities agree that for even the most steady dead load a lower factor of safety than 3 must not be used.

Besides the kind of load, the reliability of the material used affects the factor of safety. Thus, a higher factor of safety is always used for cast iron than for steel and wrought iron, and higher still for wood and brickwork. Furthermore, the conditions as to wear, rust, etc., must be taken into consideration. It can thus be seen that the factor of safety to be used varies considerably. The only thing that can be said in general is, that in ordinary practice the factor of safety ranges from 3 to 20;

if shocks in the machinery are of common occurrence, 24 should be used.

Shafting.

Shafts are subjected to two kinds of stresses, the transverse or bending stress, and the torsional or twisting stress. Both depend on the material of which the shaft is made, and the diameter of the shaft; the bending stress also depends on the length of the shaft, i. e., the distance between the bearings. The bending stress can in many cases be almost eliminated by supporting the shaft with bearings at short distances, and placing every pulley, gear, etc., very close to a bearing.

Horse-power Transmitted and Diameter of Shaft.—Various formulas are given for these calculations, based generally on a factor of safety of from 10 to 20. In the following formulas, H. P. = horse-power safely transmitted, d = diameter of shaft in inches, r = revolutions of shaft per minute.

Transmission without pulleys, and consequently without bending stress:

Turned iron shafts. $\text{H. P.} = \frac{d^3 \times r}{50}$

$$d = \sqrt[3]{\frac{\text{H. P.} \times 50}{r}}$$

Cold rolled shafts. $\text{H. P.} = \frac{d^3 \times r}{30}$

$$d = \sqrt[3]{\frac{\text{H. P.} \times 30}{r}}$$

Line shafting with pulleys, hangers 8 feet apart:

Turned iron shafts. $\text{H. P.} = \frac{d^3 \times r}{90}$

$$d = \sqrt[3]{\frac{\text{H. P.} \times 90}{r}}$$

Cold rolled shafts. $\text{H. P.} = \frac{d^3 \times r}{55}$

$$d = \sqrt[3]{\frac{\text{H. P.} \times 55}{r}}$$

Head shafts, with bearings close to pulleys or gears:

Turned iron shafts. $\text{H. P.} = \frac{d^3 \times r}{125}$

$$d = \sqrt[3]{\frac{H.P. \times 125}{r}}$$

Cold rolled shafts. $H.P. = \frac{d^3 \times r}{75}$

$$d = \sqrt[3]{\frac{H.P. \times 75}{r}}$$

Example.—How many horse-power can safely be transmitted by a cold rolled line shaft 2-7/16 inches in diameter, making 200 revolutions per minute?

Solution.—According to the formula, we have:

$$H.P. = \frac{2\text{-}7/16 \times 2\text{-}7/16 \times 2\text{-}7/16 \times 200}{55} = 52.6.$$

Example.—What must be the diameter of a turned iron head shaft, which should transmit 170 H. P. at 175 revolutions per minute?

Solution.—$d = \sqrt[3]{\frac{170 \times 125}{175}} = 4.95$ inches.

A shaft of 4-15/16 inches diameter must be used.

Cold Rolled Shafting.—It will be seen from the above formulas that the factor of safety is always smaller where the calculation applies to cold rolled shafting. This is because such shafting resists stresses much more uniformly than does the untreated metal. It is made exactly to gauge, and for general purposes requires no further preparation. The metal is much more homogeneous than common iron.

R. H. Thurston has made a close investigation of the properties of cold rolled shafting, and has shown that the tensile strength is increased 20 to 45 per cent, and the elastic limit 15 to 97 per cent. In transverse tests, the resistance is increased 11 to 30 per cent at the elastic limit, and 13 to 69 per cent at the yield point. In torsion, the resistance at the yield point increased 31 to 64 per cent, and at the point of fracture it decreased 4 to 10 per cent.

Frictional Resistance of Shafting.—Morin and Weber determined the frictional resistance of shafting, and found the amount of work in foot-pounds per revolution lost on this account, by horizontal shafting running in bearings with continuous oiling, to be:

$$0.0112 \times P \times d.$$

P = total weight of shafting plus stress of belts in pounds.
d = diameter of shaft in inches.

The formula for the horse-power consumed by frictional resistance will then be:

$$\text{H. P.} = \frac{0.0112 \times P \times d \times r}{33{,}000}$$

r = number of revolutions per minute.

From these formulas it can be seen that the tighter the belts have to be the more power is lost by friction. The resistance of upright shafting is somewhat smaller, amounting to about 75 per cent of that of horizontal shafts.

Lining and Leveling Shafting.—After a shaft has been put up perfectly level, care must be taken to have it parallel to

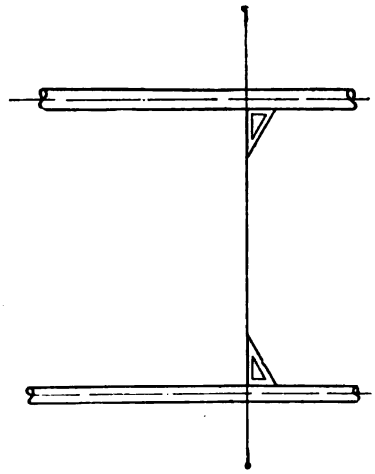


Figure 140. Lining and Leveling Shafting.

the shaft of the engine or motor. For this purpose a thin string is stretched across the two shafts in such a manner that it is absolutely perpendicular to the engine shaft, which can be accomplished by means of a square held against the main shaft. After the string has been tightly fastened in the proper position, the square is held against the new shaft to be lined, and the latter cautiously moved until the side of the square which stands perpendicular on the shaft is exactly parallel to the string (see Figure 140).

Hangers.

Hangers are devices for supporting shafting from overhead or from the side. The hangers mostly used at the present time are of the *ball and socket type,* as illustrated in Figure 141.

The two half boxes are provided with turned spherical surfaces made to fit into concave surfaces in two plungers. The plungers are provided with a coarse thread and fit into the frame of the hanger. They are kept from turning by two set-screws. The plungers thus permit of vertical adjustment while the ball and socket aligns itself with the shaft. The possibility of vertical adjustment as afforded by this type of bearing is of the greatest importance, since it is only necessary to put up the hangers with the centers of the boxes in the same vertical plane.

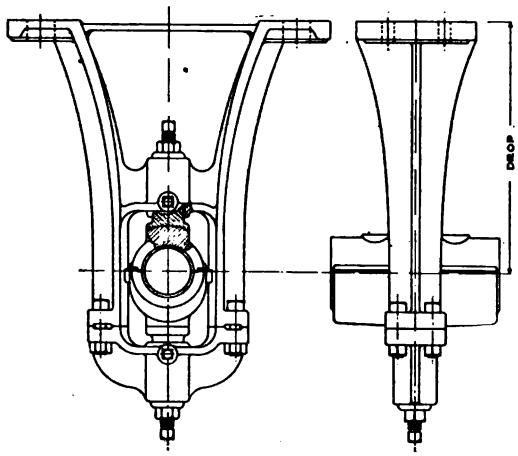


Figure 141. Ball and Socket Hanger Bearing.

Any adjustment necessary up or down is readily made by means of the plungers.

The distance from the place where the hanger is fastened against the ceiling to the center of the box is called the *drop.* When ordering hangers it is necessary to mention the diameter of the shaft and the drop in inches, as, for instance, a $2^{15}/_{16}'' \times 18''$ drop hanger.

In case the hanger to be used is a *post hanger,* then the distance from wall or post to the center of the box is the drop. These hangers are often provided with concave feet for fasten-

ing to cast iron columns. All hangers can be had with belt shifter bar cast on.

Shaft Bearings.

Shaft bearings are either made of brass or of babbitt metal. The former should be employed only where the pressure is excessive, where sudden jars occur, or where unusually high speeds are required. For ordinary conditions babbitt metal bearings are preferable and offer a number of advantages.

Advantages of Babbitt Bearings.—Babbitt bearings can be more easily kept cool than brass bearings, and there is less liability of the journal cutting into the shaft. Babbitt metal gives less friction, and on account of its softness will accommodate itself to any slight irregularities in the surface of the journal. Should the shaft get slightly out of alignment, the metal will yield where the pressure is greatest, until there is a uniform bearing surface. The wear of the shaft will also be longer, and when the bearings become worn they can be rebabbitted at little expense.

In order to obtain most satisfactory results, babbitt metal of the proper composition, and consequently of the proper hardness, should be selected for each individual case.

Pressure on Bearings.—For continuous running, the pressure on bearings should not exceed 180 pounds per square inch of bearing surface, according to the author's experience, and if feasible, the pressure should preferably not be higher than from 100 to 150 pounds for common bearings.

Dimensions of Bearings.—The actual bearing surface can be calculated by multiplying the diameter of the shaft by the length of the journal. The proper length of the latter can easily be found if the horse-power to be transmitted is known, as may be seen from the following example.

Example.—A 5-inch shaft, making 175 revolutions per minute, over a pulley 60 inches in diameter, is to transmit 170 H. P., and the pressure on the bearings should not exceed 150 pounds per square inch. What should be the length of the bearing?

Solution.—Change the horse-power to be transmitted into foot-pounds, i. e., $170 \times 33{,}000 = 5{,}610{,}000$ foot-pounds. Cal-

culate the number of feet a point on the surface of the shaft travels per minute, which amounts to $\frac{5 \times 3.1416 \times 175}{12} = 229$ feet. Dividing the total number of foot-pounds by this quantity, we obtain: $\frac{5{,}610.100}{229} = 24{,}500$ pounds, approximately, which is the dead weight resting on the bearings.

We wish to allow 150 pounds per square inch of bearing surface or $150 \times 5 = 750$ pounds around the shaft for one inch length of the bearing. Therefore, by dividing the dead weight 24,500 pounds by 750 we obtain the length of the bearings: $\frac{24{,}500}{750} = 32.7$ inches.

As we have one bearing on each side of the pulley, the length of each one must be $\frac{32.7}{2} = 16.35$, or in round numbers, 17 inches.

Self-Oiling Bearings.—Bearings with oil chambers underneath, so-called self-oiling bearings, are preferable to those that must be oiled from the top.

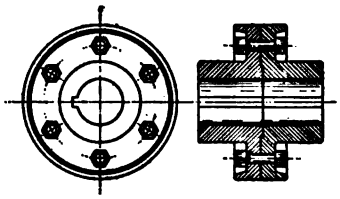


Figure 142. Flange Coupling.

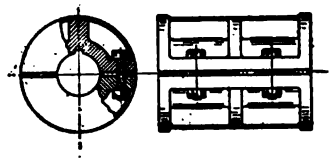


Figure 143. Split Sleeve Coupling.

Shaft Couplings.

There are two kinds of shaft couplings in general use, the *flange* and the *sleeve* coupling, which are illustrated in Figures 142 and 143, respectively.

The flange coupling should be faced off after it has been keyed on the shaft, and should be secured by close fitting bolts in reamed holes, in order to possess the desired strength and rigidity.

In the latter respect the sleeve couplings are inferior; still they are preferred by some engineers, for the reason that they are compelled to employ open-sided hangers with the flange

couplings, or, if closed hangers are used, the lower part must be removable. Furthermore, the pulleys must be of the split type, in order to avoid difficulties when the shaft has to be taken down or pulleys removed. Still, as mentioned before, a sleeve coupling is never as rigid as a flange coupling.

So-called *special couplings* are sometimes employed for connecting shafts whose axes do not form a continuous straight line. In this manner a shaft may be operated which runs at an angle of almost 15°. If the angle is to be wider, a double joint must be used. Figure 144 shows a coupling of this kind.

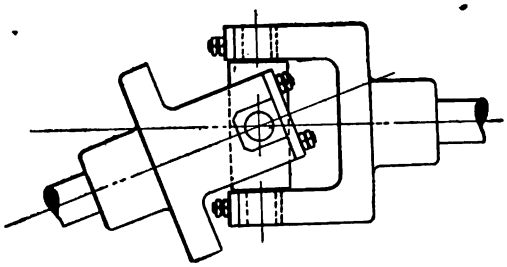


Figure 144. Special Coupling.

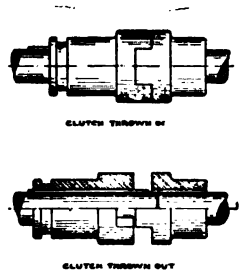


Figure 145. Jaw Clutch.

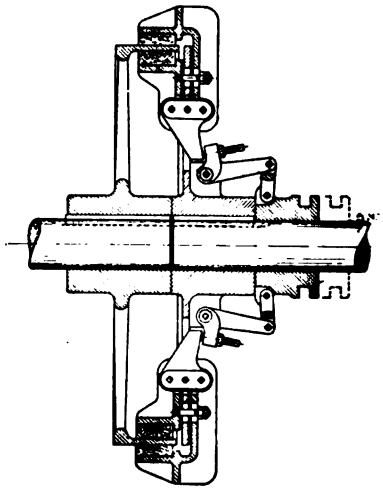


Figure 146. Clutch Coupling.

Clutches.

A clutch is a device for connecting one shaft with another by simply moving a lever. There are two types of clutches, the so-called jaw clutch and the friction clutch.

Jaw Clutch.—This clutch causes a positive engagement by interlocking the connected parts, but is only used for light work, where the rotating shaft can be stopped to make the junction. Figure 145 illustrates the jaw-clutch.

Friction Clutch.—This device, which may be a clutch coupling or a clutch pulley, permits connecting the rotating shaft with the shaft at rest without stopping the former, or it may

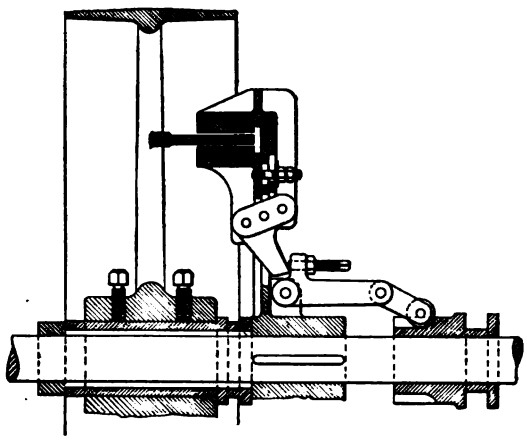


Figure 147. Clutch Pulley.

connect a counter-shaft with the main shaft, as described below. In order to prevent the shock due to sudden starting of the shaft or the machinery at rest, the clutch does a certain amount of slipping when connection is being made.

If a clutch simply connects two shafts with each other forming a straight line, it is called a *clutch coupling,* but if it connects a countershaft with the rotating shaft it is designated by the name of *clutch pulley.* In the latter case the clutch grasps the loose pulley on the shaft and sets it in motion, and, by the belt running from this pulley to the one on the countershaft,

the latter is also set in motion. Figure 146 illustrates a clutch coupling, and Figure 147 illustrates a clutch pulley.

Keys.

Keys are pieces of steel employed for securing wheels, pulleys, clutches, couplings, rolls, etc., to a shaft. There are four kinds of keys, the sunk key, the feather key, the flat key, and the hollow key. The construction of these keys is shown in Figure 148.

Sunk Key.—The sunk key is the one most commonly used. It is employed in all cases where the strain is great. It is tapered and wedged in tightly between the hub and the shaft.

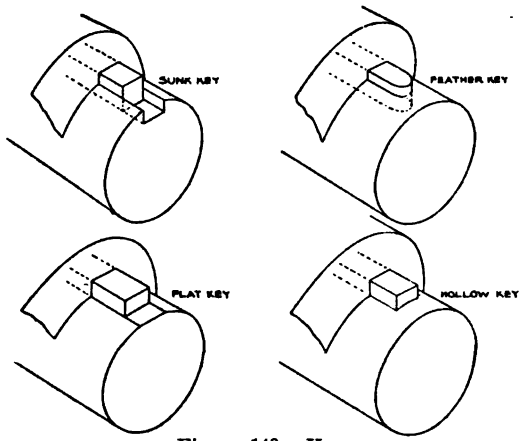


Figure 148. Keys.

Feather Key.—The feather key is employed where it is desired to have the wheel slide along the shaft, the key-way extending on the shaft beyond the pulley or wheel. It is not tapered. Usually the key is of the same length as the key-seat, and is fastened to the shaft. The pulley then slides over the shaft and the key.

Flat Key.—This key beds upon a flat place on the shaft, and is used where it is necessary to secure the wheel more firmly without weakening the shaft by cutting a key-way into it.

Hollow Key.—The hollow key is employed where the wheel may have to be moved to a different place on the shaft, and where it is undesirable that the latter have any flat place on it or a recess cut into it.

Dimensions of Sunk Keys, in Inches.

Diam. of Shaft...	1	1¼	1½	1¾	2	2½	3	3½	4	4½	5	6
Breadth of Key..	5/32	7/32	1/4	11/32	13/32	15/32	17/32	9/16	11/16	13/16	7/8	15/16
Depth of Key....	3/16	1/4	5/16	3/8	7/16	1/2	5/8	5/8	3/4	7/8	15/16	1

Dimensions of Feather Keys, in Inches.

Diam. of Shaft...	1¼	1½	1¾	2	2½	2½	3	3½	4	4½
Breadth of Key..	1/4	1/4	5/16	5/16	3/8	3/8	1/2	9/16	9/16	5/8
Depth of Key....	3/8	3/8	7/16	7/16	1/2	1/2	5/8	3/4	3/4	7/8

Dimensions of Flat Keys, in Inches.

Diam. of Shaft...	1	1¼	1½	1¾	2	2½	3	3½	4	5	6	7	8
Breadth of Key..	1/4	5/16	3/8	7/16	1/2	5/8	3/4	7/8	1	1 1/8	1 3/8	1 1/2	1 3/4
Depth of Key....	5/32	5/16	1/4	9/32	5/16	3/8	7/16	1/2	5/8	11/16	13/16	7/8	1

Pulleys.

Belt Pulleys.—The materials from which belt pulleys are made are cast iron, sheet steel, and wood. Figure 149 shows a common cast iron pulley, and Figure 150 a wood split pulley.

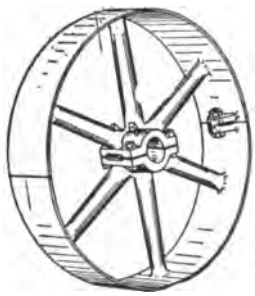


Figure 149. Cast Iron Split Pulley.

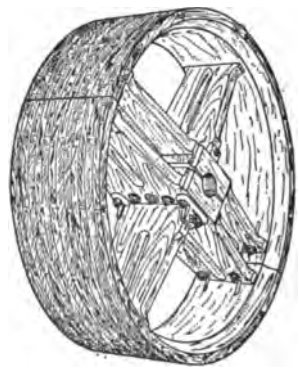


Figure 150. Wood Split Pulley.

Split Pulleys are very convenient, and should be used wherever possible, since they can be taken off without interfering with any other pulley or with the shaft.

The face of a pulley is either made *crowned* (convex) or *straight*.

Step Cone Pulleys.—These pulleys, illustrated by Figure 151, are commonly used in machine shops. They permit regulating the speed of the machinery, the various speeds obtainable

being fixed in advance by the size of the pulleys of which the device consists.

Cone Pulleys.—These also permit regulating the speed of the machinery; the speeds, however, are not fixed, but can be changed at will by simply shifting the belt to the corresponding place on the pulleys. These pulleys, which are shown in Figure 152, are employed in special cases only.

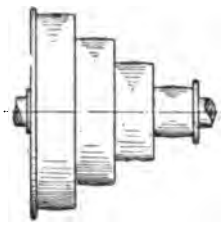


Figure 151. Step Cone Pulley.

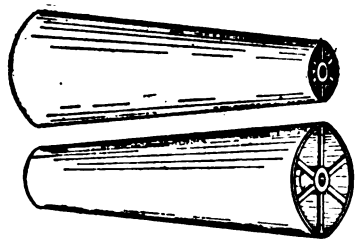


Figure 152. Cone Pulleys.

Tight and Loose Pulleys.—These are generally employed for starting or stopping a certain part of the machinery. Shifting forks, placed near the driven pulley, shift the belt from the loose to the tight pulley if the machine is to be started, the belt adjusting itself on the driver. By moving the belt from the tight to the loose pulley the machinery is stopped. Figure 153 shows a tight and loose pulley.

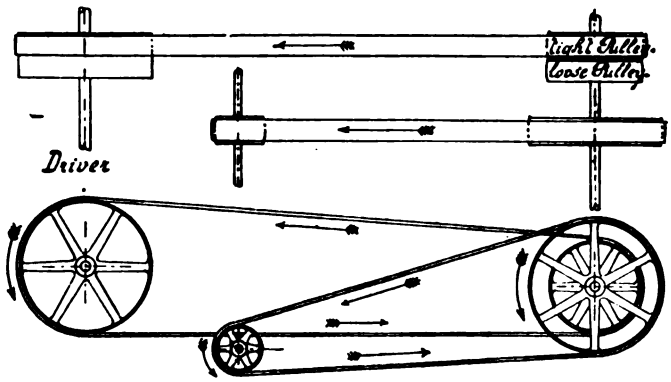


Figure 153. Tight and Loose Pulley.

It is essential that loose pulleys be well lubricated. For this purpose they are generally provided with oil holes and

grooves for distributing the oil. It is preferable, however, to have a reservoir for the lubricant cored out in the hub.

Covered Pulley.—In order to increase the adhesion of the belt on the pulleys, the latter are sometimes covered with paper, leather, or rubber. Such pulleys transmit from 20% to 50% more power.

Balance of Pulley.—In order to run evenly the pulley must be well balanced, i. e., the center of gravity of the pulley must be in the axis of the shaft. A pulley which is not well balanced is liable to cause a knock in its bearings when it revolves rapidly, since that part which is out of balance tries to follow a circle of revolution other than that described by the other parts of the rim.

Size and Speed of Pulleys.—For calculating the diameter and speed of pulleys the following simple formulas are used. They are all derived by transposition from the formula:

$$D \times R = d \times r.$$

$$D = \frac{d \times r}{R}$$

$$d = \frac{D \times R}{r}$$

$$R = \frac{d \times r}{D}$$

$$r = \frac{D \times R}{d}$$

D = diameter in inches of driving pulley or driver.

d = diameter in inches of driven pulley.

R = revolutions per minute of driving pulley.

r = revolutions per minute of driven pulley.

Example.—What must be the diameter of the driving pulley if it makes 250 revolutions per minute, and if the diameter of the driven is 12 inches and is to make 500 revolutions per minute?

Solution.—$D = \frac{d \times r}{R} = \frac{12 \times 500}{250} = 24$ inches.

Example.—The diameter of the driver is 24 inches and it makes 250 revolutions per minute. How many revolutions per minute does the driven make, if its diameter is 12 inches?

Solution.—$r = \frac{D \times R}{d} = \frac{24 \times 250}{12} = 500$ revolutions.

If the power is to be transmitted over a number of countershafts, the calculation of size and speed of the pulleys is similar. The diameter of the driven pulley is found by multiplying together the diameters of all the drivers and the speed of the first driver, and dividing the result by the product of the diameters of the driven and the speed of the last driven. Expressed in a formula, the calculation reads as follows:

$$d_n = \frac{D_1 \times D_2 \times D_3 \times \ldots\ldots \times R}{d_1 \times d_2 \times \ldots\ldots \times r}$$

D_1 = diameter in inches of first driver.
D_2 = diameter in inches of second driver.
D_3 = diameter in inches of third driver.
R = revolutions per minute of first driver.
d_1 = diameter in inches of first driven.
d_2 = diameter in inches of second driven.
d_n = diameter in inches of last driven.
r = revolutions per minute of last driven.

Example.—What must be the diameter of a pulley for a machine which is to make 1,800 revolutions per minute? The power is to be transmitted over three shafts. The engine makes 60 revolutions per minute, and has a belt wheel of 10 feet diameter, which drives a line shaft by a pulley of 4 feet diameter. From this line shaft a countershaft is driven by a pair of pulleys, of which the driver on the line shaft is 48 inches in diameter and the driven on the countershaft 24 inches. The driver on the countershaft is 42 inches in diameter.

Solution.—$d_n = \frac{120 \times 48 \times 42 \times 60}{48 \times 24 \times 1{,}800} = 7.$

The diameter of the last driven must be 7 inches.

As a rule, the smaller pulley should have at least one-third the diameter of the large one, in order to obtain a sufficiently large arc of contact.

Width of Pulleys.—Pulleys should be made about 20% wider than the belt driving them.

Distance between Pulleys.—The most satisfactory results are obtained if the distance between pulleys is from 20 to 25 feet from center to center. It should never be less than 2½ times the diameter of the large pulley.

Idler Pulleys.—These pulleys are not used for transmitting power, but only for carrying the sag in a long belt or for acting as guides when the direction of a transmission is to be changed. They are therefore called *idlers*. If not properly adjusted they may cause a belt to run to one side.

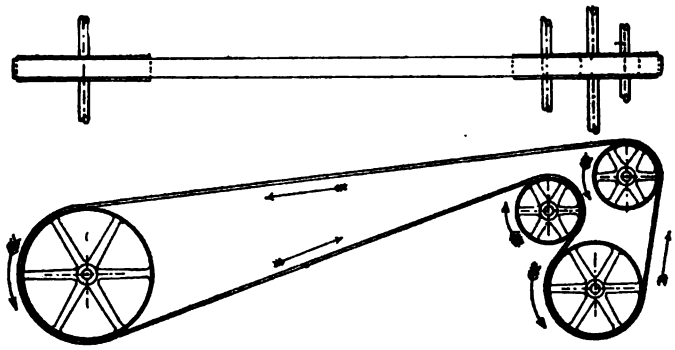


Figure 154. Idler Pulleys.

Tightening Pulleys.—These are idlers mounted on a movable carriage. They are used for tightening long belts, especially

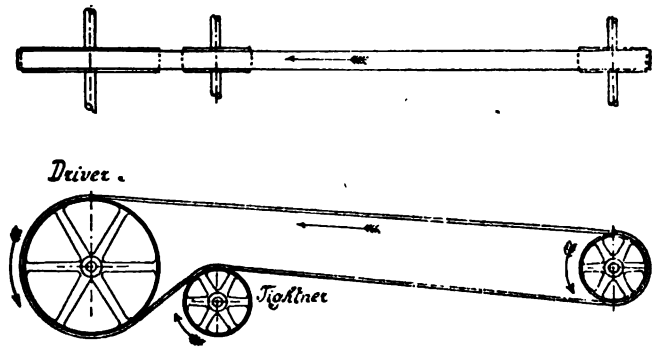


Figure 155. Belt Tightener.

vertical ones, and have the advantage that they make a clutch superfluous, inasmuch as they cause the belt to be set in motion or to be put at rest.

Tighteners are also frequently used on main driving belts to increase the arc of contact on the smaller pulley. They work most satisfactorily when placed on the slack side of the belt at

about one-fourth of the distance between shafts from the driving pulley. Figure 155 illustrates a tightener.

Friction Pulleys.—Friction pulleys are devices by which power is transmitted from one shaft to another by drawing one pulley tight against the other. In order to increase the friction between the two pulleys, one is frequently made of paper or wood and the other of cast iron.

Figure 156. Friction Pulley.

The loss of power due to the excessive pressure created in the bearings is greater in this kind of power transmission than with any other, and for this reason friction pulleys are only used in special cases where an absolute and quick stop is required, as, for instance, in hoisting machinery. Figure 156 illustrates a set of friction pulleys.

Belts.

In order to transmit power from one shaft to another, strips of flexible material, called belts, are stretched over two pulleys, one pulley acting as the driver and the other as the driven.

Material for Belts.—Leather and rubber are generally employed for belting. The latter material is cheaper and superior to leather, if the belt is exposed to dampness, which soon makes a leather belt worthless. Rubber belts also transmit at least 25% more power, but their life is considerably shorter.

A leather belt will wear more satisfactorily and transmit more power if run with the finished side next to the pulleys, because a better contact with the pulley and consequently more friction between the belt and pulley is obtained. A leather belt will stretch about 2 per cent of its length during the first six months; during its lifetime a belt may stretch more than 6 per cent. In general, belts are more durable and work better if made narrow and thick, than if wide and thin.

Friction between Belt and Pulley.—The friction between a belt and a pulley depends upon the material, the condition of belt and pulley, the tension of the belt, and the arc of contact, but is independent of the area of the surface of contact.

Calculation of Width of Belt and Horse-Power Transmitted.—The main factors determining the amount of power that can be transmitted by a belt are: The speed of the belt in feet per minute, the arcs of contact, the position in which the belt travels, the weight of the belt, the thickness, the width, and the amount of adhesion between belt and pulley.

The best speed of belts is from 4,000 to 4,500 feet per minute, and when installing transmission machinery the pulleys should be so arranged, if possible, that their rims move with this velocity.

There are several rules according to which the transmitting capacity of a belt can be calculated. Some formulas for calculating horse-power transmitted, diameter of pulley, number of revolutions, and width of belt, are given below:

For Single Belts.—

$$\text{H. P.} = \frac{D \times N \times w}{2{,}750}$$

$$D = \frac{2{,}750 \times \text{H. P.}}{w \times N}$$

$$N = \frac{2{,}750 \times \text{H. P.}}{w \times D}$$

$$w = \frac{2{,}750 \times \text{H. P.}}{D \times N}$$

For Double Belts.—

$$\text{H. P.} = \frac{D \times N \times W}{1{,}925}$$

$$D = \frac{1{,}925 \times \text{H.P.}}{W \times N}$$

$$N = \frac{1{,}925 \times \text{H.P.}}{W \times D}$$

$$W = \frac{1{,}925 \times \text{H.P.}}{D \times N}$$

H. P. = horse-power which can be transmitted.
D = diameter of pulley in inches.
N = number of revolutions of pulley per minute.

w = width of single leather belt in inches.
W = width of double leather belt in inches.

Instead of the factors 2,750 and 1,925, other factors are sometimes employed, viz., 3,300 for single and 2,100 for double leather belts.

Example.—How many H. P. can a single leather belt 9 inches wide transmit over a pulley which has a diameter of 40 inches and makes 350 revolutions per minute?

Solution.—H. P. $= \frac{40 \times 350 \times 9}{2{,}750} = 45.8.$

Example.—What must be the width of a double leather belt if 50 H. P. are to be transmitted over a receiving pulley 42 inches in diameter, making 300 revolutions per minute?

Solution.—W $= \frac{1{,}925 \times 50}{42 \times 300} = 8$ inches approximately.

Calculations from Belt Speed.—Another rule easily remembered and frequently applied is to allow for single leather belts 800 feet per minute of belt speed and for double leather belts 550 feet per minute of belt speed, for transmitting one H. P. per inch of belt width. Some allow 1,000 feet for double leather, and 600 feet for single leather.

Example.—What must be the width of a double leather belt, if 50 H. P. are to be transmitted over a receiving pulley 36 inches (3 feet) in diameter, making 350 revolutions per minute?

Solution.—The width of the belt must be:

$$\frac{50 \times 550}{3 \times 3.1416 \times 350} = 8 \text{ inches approximately.}$$

If there is a considerable difference between the diameters of the two pulleys, the arc of contact of the smaller pulley is reduced, and a 9-inch belt would give better results.

Example.—How many H. P. can a single leather belt 10 inches wide transmit, which runs over a 42-inch (3.5 feet) pulley, making 300 revolutions per minute?

Solution.—The belt can transmit:

$$\frac{10 \times 3.5 \times 3.1416 \times 300}{800} = 41.23 \text{ H. P.}$$

Calculation of H. P. from Stress.—The number of H. P. transmitted by a belt can also be found by allowing a certain stress on the belt per inch of width, and in a similar manner it is

possible to calculate the width of the belt and to find the stress, if a certain number of H. P. are transmitted. The formulas used are the following:

$$\text{H. P.} = \frac{w \times D \times 3.1416 \times N \times S}{33{,}000}$$

$$w = \frac{\text{H. P.} \times 33{,}000}{D \times 3.1416 \times N \times S}$$

$$S = \frac{\text{H. P.} \times 33{,}000}{D \times 3.1416 \times N \times w}$$

H. P. = horse-power transmitted.
w = width of belt in inches.
D = diameter of pulley in feet.
N = number of revolutions per minute.
S = stress in pounds per inch of width.

On the average, a stress of from 20 to 40 pounds is allowed on single leather belts, and from 50 to 70 pounds on double leather belts, per inch of width.

Example.—What must be the width of a double leather belt for transmitting 50 H. P. over a pulley 36 inches in diameter making 300 revolutions per minute? The stress on the belt must not exceed 60 pounds per inch of width.

Solution.—According to the second formula we have:

$$w = \frac{50 \times 33{,}000 \times 12}{36 \times 3.1416 \times 300 \times 60} = 10 \text{ inches.}$$

Example.—What is the stress per inch of belt width, if a 10-inch belt transmits 40 H. P. over a pulley 42 inches in diameter, making 300 revolutions per minute?

Solution.—

$$S = \frac{40 \times 33{,}000}{3.5 \times 3.1416 \times 300 \times 10} = 40 \text{ pounds.}$$

From the formula and the example it can be seen that the stress is found by dividing the total number of foot-pounds transmitted by the number of feet the belt travels in one minute times the width of the belt in inches.

The greater the stress, the shorter the life of a belt. For instance, double belts, with a stress of 65 pounds per inch of width, running day and night will last about 7 years, while with a stress of 30 pounds they will last about 18 years. Since the

stress decreases as the belt width increases, the advantage of ample belt width is evident.

It can furthermore be seen from the foregoing formulas that the capacity of a belt for transmitting power varies in direct proportion to the width of the belt and also to its speed. If a 3-inch belt, for instance, transmits 5 H. P. at a certain speed, a 6-inch belt will transmit 10 H. P. at the same speed. If the width remains the same while the speed is doubled, the 3-inch belt will transmit 10 H. P. However, this only holds good if the speed is kept within practical limits, i. e., does not exceed 5,000 feet per minute.

Methods of Connecting Shafts by Belts.—Connection of two shafts by belts is accomplished in different manners, depending on the relative position of the shafts and on the direction of their rotation, i. e., whether they are running in the same or in opposite directions.

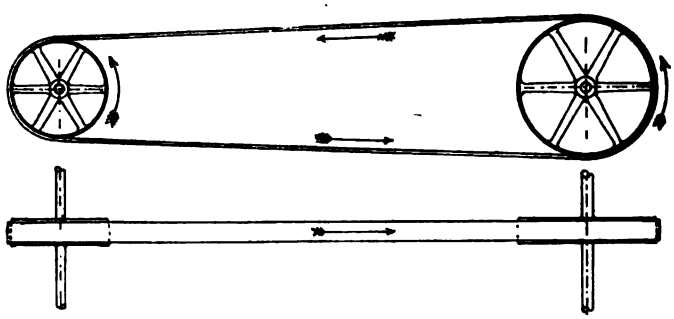


Figure 157. Open Belt.

Figure 157 represents an *open belt,* i. e., the connection of two parallel shafts running in the same direction.

In Figure 158 is illustrated a *crossed belt* connection between two parallel shafts running in opposite directions.

Figure 159 shows a *quarter twist belt.* Such a belt is employed for connecting two shafts running at right angles to each other, one a considerable height above the other. The connection is made as follows: Set the pulleys, which should be crown-faced, so that a plumb line dropped from the center of the face of the upper pulley at *c* on the side where the belt leaves it, will just touch the center of the face of the lower pulley at *a.*

If the belts are to run through floors or walls, the exact place for their passage must first be determined. For this purpose it is best to draw pulleys, shafting, floor and walls to scale on paper, and put in the belt as shown in the figures below. The position and size of the necessary holes are then easily found. It must not be overlooked, however, that sufficient allowance must be made for the sag movement of the belt when in operation.

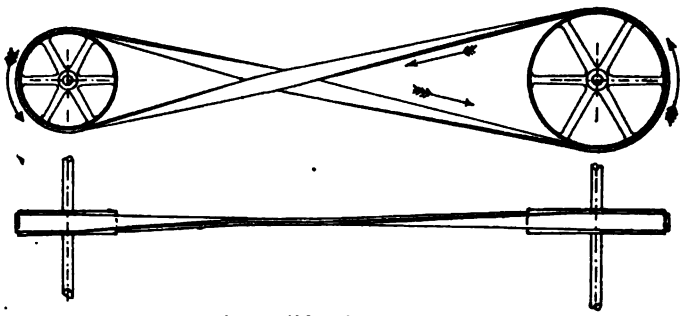


Figure 158. Crossed Belt.

In Figure 159 is shown a belt transmission around a corner between shafts at right angles to each other. It is possible to make such connections between shafts running at any desired

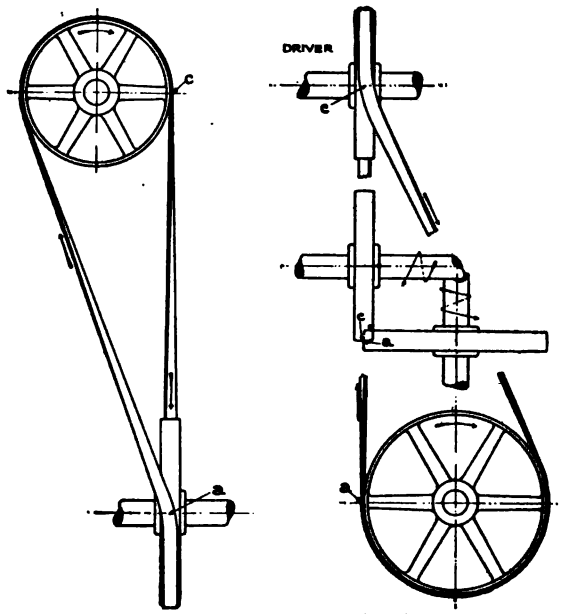


Figure 159. Quarter Twist Belt.

angle; it being only necessary to guide the belts by means of idlers, which give the belt the desired direction.

Joining of Belts.—The joining of the ends of belts may be done in three ways.

1. By lacing the ends together with rawhide or strips of tanned leather, this method being called *belt lacing.*

2. By tapering the ends of the belt and *glueing* them together.

3. By fastening the ends by means of *metal hooks* or *links.*

Lacing of Belts.—In order to lace the ends of a belt, holes must first be punched for the lacing. Oval punches are better than round ones. The size of the punch, as well as that of the lacing, varies with the width of the belt as follows:

3-inch belt, ¼-inch lace, punch No. 6 ($^{6}/_{16}$-inch diameter).

3 to 6-inch belt, ⅜-inch lace, punch No. 8 ($^{7}/_{32}$-inch diameter).

7 to 14-inch belt, ½-inch lace, punch No. 10 (¼-inch diameter).

Over 14-inch belt, ⅝-inch lace, punch No. 12 ($^{9}/_{32}$-inch diameter.

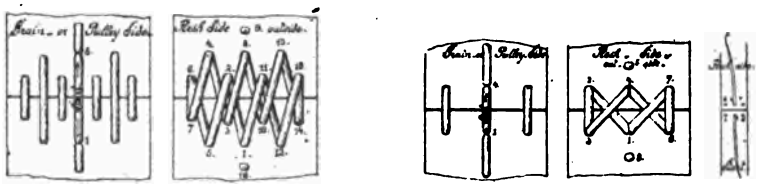


Figure 160. Belt Lacing.

Two rows of holes in zigzag should be punched in each end, the number of holes in each row being determined by the width of the belt:

3-inch belt 2 holes in each row.
6-inch belt 4 holes in first row,
3 holes in second row.
10-inch belt 5 holes in first row,
4 holes in second row.

The edges of the holes should be ¾ inch from the sides and ⅞ inch from the ends.

Figure 160 illustrates how a belt should be laced. It will be perceived that the flesh side is the outside. This should always be the case, since the grain (hair) side of the belt is

smoother than the flesh side and therefore adheres better to the pulley, and since, on the other hand, the flesh side possesses more elasticity and is consequently less liable to crack by stretching while passing over the pulleys.

Glueing Belts.—The belts are cut from 6 to 12 inches longer than necessary and the ends tapered or cut in zigzag form and then glued together. In this manner the joined ends have the same thickness as the rest of the belt.

Belts of this kind are generally called lapped or spliced belts. They have the advantage of running with less noise than belts joined in a different manner, since there is no lacing to interfere with the pulley. They are usually employed as main driving belts and for running dynamos and motors or wherever the stretch of the belt can be taken up by a tightener or by shifting the machine.

Joining by Belt Hooks.—Belt hooks are made in various forms as shown in Figure 161.

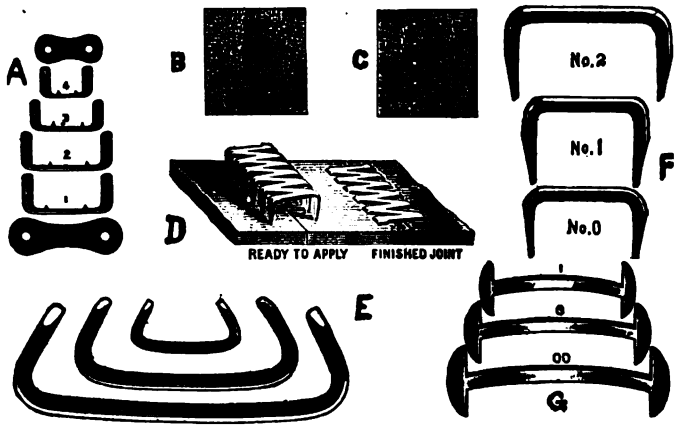


Figure 161. Belt Hooks.

A illustrates what may be called a double rivet link. In B and C are shown how the finished side and the flesh side of the belt look after joining by this hook.

In D is shown another form of hook which can readily be applied by simply driving down with a hammer and clinching the points projecting on the other side of the belt. E and F are hooks which are also clinched or pounded flat after the belt has been joined by them.

When using Blake's belt studs illustrated in G, slots must be cut parallel to the sides of the belt and the studs inserted in the same manner as cuff buttons are put into a cuff.

The use of belt hooks has the advantage that a belt can be joined much quicker than with lacing. They are therefore frequently used.

Determining Length of Belting.— Whenever feasible, the proper length of belting is best determined by direct measurement with a tape line around the two pulleys. From the length thus found, ¼ inch should be subtracted for every foot, which will give the belt the proper tension.

To calculate the length of belts accurately is somewhat complicated, requiring the use of trigonometrical functions. The formulas below are much simpler, and give sufficiently accurate results.

Open Belts.—For open belts, find the average circumference of the two pulleys (add the two diameters, divide by two and multiply by 3.1416), and add to this twice the distance between the centers of the shafts where the pulleys are of nearly the same diameter. This method gives good results; however, more accurate results may be obtained by making a small addition as shown below. The formula follows:

$$L = \frac{d_1 + d_2}{2} \times 3.1416 + 2 \times D + \frac{(d_1 - d_2) \times (d_1 - d_2)}{4D}$$

L = length of belt in feet.
d_1 = diameter of first pulley in feet.
d_2 = diameter of second pulley in feet.
D = distance between shaft centers in feet.

Example.—What must be the length of an open belt which is to run over a pair of pulleys 4 feet and 3 feet in diameter respectively? The distance between the two shafts is 18 feet from center to center.

Solution.—$L = \frac{4+3}{2} \times 3.1416 + 2 \times 18 + \frac{1}{16} = 47$ feet.

From this result must be subtracted a certain amount to allow for tension; this is ¼ inch per foot for single belts, and ⅛ to $\frac{3}{16}$ inch for double belts.

Crossed Belts.—For crossed belts, the calculation is a little more complicated. The formula reads as follows:

$$L = \frac{d_1 + d_2}{2} \times 3.1416 + 2 \times \sqrt{\frac{(d_1 + d_2) \times (d_1 + d_2)}{4} + D \times D}.$$

Example.—What must be the length of a crossed belt running over a pair of pulleys 4 and 3 feet in diameter, if the distance between the two shafts is 18 feet?

Solution.—

$$L = \frac{4+3}{2} \times 3.1416 + 2 \times \sqrt{\frac{(4+3) \times (4+3)}{4} + 18 \times 18}$$

$=47.7$ feet.

The same allowance for tension must be made as above, so that for single leather belting the final length would be $47.7 - \frac{47.7}{4 \times 12}$ or $47.7 - 1$ or 46.7 feet.

Quarter Twist Belts.—The calculation is similar to the above.

$$L = \frac{d_1 + d_2}{2} \times 3.1416 + D + \sqrt{\frac{(d_1 + d_2) \times (d_1 + d_2)}{4} + D \times D}.$$

Example.—What must be the length of the belt under the conditions of the preceding example?

Solution.—

$$L = \frac{4+3}{2} \times 3.1416 + 18 + \sqrt{\frac{(4+3) \times (4+3)}{4} + 18 \times 18}$$

$=47.33$ feet.

Allowance must of course be made for tension, as above.

Chains or Link Belting.

Where the distance between shafts is not great enough to use belting, but too great for spur gears, and where a positive motion is wanted, for heavy loads and under conditions where other belting would not answer, as under water or in very hot places, so-called link belting is employed.

This device for transmitting power from one shaft to another, consists of a chain running over two wheels called sprockets, which have comparatively large teeth. The chains are generally made of malleable iron, and one link is hooked into the other.

For very heavy loads, they are made of wrought iron, and consist of short rods and cross bars, or of flat pieces of iron through which a pin passes at each end.

The wear and tear of detachable link belting running over sprocket wheels is considerable; they are also very noisy and limited in speed.

A few years ago a chain of the form shown in Figure 162

Figure 162. Silent Chain.

was introduced. With it comparatively high speed, up to 1,400 feet per minute, has been obtained. At the same time there is less noise than with an ordinary belt. These chains are called

Figure 163. Peerless Noiseless Chain.

silent chains. For transmitting heavy loads over a short distance they have given excellent satisfaction.

The driving chain illustrated in Figure 163 consists of flat steel center links and bevel-shaped side links, and is encased bot-

tom and sides by a strip of canvas belting, the side faces of which present friction surfaces which run in driving contact with the sides of the taper grooves in the sheave. The chain runs in a V-shaped pulley and as there is no metal contact, either on chain or pulley, it is noiseless. It operates without lubrication, although it is constructed so that it can be oiled if desired, which, of course, will tend to greatly increase its life.

Ropes.

Rope transmission offers many advantages over other methods of transmitting power from shaft to shaft, and it has come into very common use.

Arrangement of Ropes.—In the English system of rope drive, each rope is separate, and if one should break, the work can usually be done by the remaining ropes. According to the American method, the rope is continuous, being wound spirally around the pulleys. The last wrap is carried around a tension pulley and from there back to the first groove of either one of the pulleys. The bottom side is usually the driving side, because in this manner the arc of contact on the driven pulley is increased by the sag of the slack side. The tension pulley is mounted on a carriage which is loaded so as to maintain a given tension on the rope, while being free to take up the stretch in the ropes.

Rope carriers should be put on the slack side of the rope and should be set in such a manner that they will not create a reverse bend in the rope.

Size and number of ropes are points of importance, which should be carefully considered in each case. It must not be overlooked that in small ropes there is less internal friction, which is the principal destructive agent, and, therefore, they possess greater efficiency and durability.

Pulleys.—The pulleys for rope transmission, generally called *sheaves,* are grooved, the sides of the grooves meeting at an angle of 45°. The grooves must be made in such a way that the rope does not lie on the bottom of it.

The size of the pulleys is of considerable importance. The diameter should not be smaller than 40 times the diameter of the rope. Sometimes the grooves of the sheaves are lined with rubber, and the wear and tear on the rope is thus considerably reduced.

Power Transmitted by Ropes.—The horse-power transmitted by ropes depends on the number of ropes used, the diameter of the sheaves, the number of revolutions and the breaking stress of the rope. It is best not to take the safe working stress any higher than from 1¼ to 1½% of the breaking stress.

For a one-inch manila rope, the latter amounts to 8,000 pounds, as can be seen from the above table. Rope drives installed with such a safe working stress will give no difficulties and will outlast a leather belt. In many cases, however, the contractor who instals the rope transmission bases his calculations on a safety factor of from 4 to 5%, and even higher, especially when trying to compete with others; the result is that the ropes stretch continuously, and their life is seldom more than 6 years.

For steel ropes, a lower factor of safety may be allowed than for manila ropes. The safe working load may well be taken as 10% of the breaking stress in this case.

Strength and Weight of Manila Rope.

Diameter.	Circumference.	Weight per Foot.	Breaking Load.	Diameter.	Circumference.	Weight per Foot.	Breaking Load.
ins.	ins.	lbs.	lbs.	ins.	ins.	lbs.	lbs.
0.239	¾	.019	560	1.92	6	1.19	25,536
0.318	1	.033	784	2.07	6½	1.39	29,120
0.477	1½	.074	1,568	2.23	7	1.62	32,704
0.636	2	.132	2,733	2.39	7½	1.86	36,288
0.795	2½	.206	4,278	2.55	8	2.11	39,872
0.955	3	.297	6,115	2.86	9	2.67	47,040
1.11	3½	.404	8,534	3.18	10	3.30	54,208
1.27	4	.528	11,558	3.50	11	3.99	61,376
1.43	4½	.668	14,784	3.82	12	4.75	68,544
1.59	5	.825	18,368	4.14	13	5.58	75,712
1.75	5½	.998	21,952	4.45	14	6.47	82,880

The horse-power transmitted, the number of ropes to be used and the number of revolutions the sheave has to make per minute can be calculated according to the following formulas:

$$\text{H. P.} = \frac{d \times 3.1416 \times r \times n \times T \times t}{33{,}000}$$

$$n = \frac{\text{H. P.} \times 33{,}000}{d \times 3.1416 \times r \times T \times t}$$

$$r = \frac{\text{H. P.} \times 33{,}000}{d \times 3.1416 \times n \times T \times t}$$

d = diameter of sheave in feet.

r = revolutions per minute of sheave.

n = number of ropes.

T = breaking stress in pounds for one rope.

t = safety factor for one rope in per cent, usually 1.5%.

Example.—How many H. P. can be transmitted by nine manila ropes, each of one inch diameter, running over a sheave 42 inches (3.5 feet) in diameter? The sheave makes 250 revolutions per minute. The breaking stress of a one-inch manila rope is 8,000 pounds; the safe working stress is 1.5% of this.

Solution.—

$$\text{H. P.} = \frac{3.5 \times 3.1416 \times 250 \times 9 \times 8{,}000 \times 1.5}{33{,}000 \times 100} = 90.$$

Example.—How many one-inch ropes are necessary for transmitting 75 H. P. over a sheave 3.5 feet in diameter, making 270 revolutions per minute?

Solution.—

$$n = \frac{75 \times 33{,}000 \times 100}{3.5 \times 3.1416 \times 270 \times 8{,}000 \times 1.5} = 7 \text{ ropes.}$$

Horse-power Transmitted by Manila Rope.

Diameter of Rope Inches.	Speed of Rope in Feet per Minute.									
	1500	2000	2500	3000	3500	4000	4500	5000	6000	7000
¾	2.9	4.0	4.8	5.5	6.3	6.9	7.4	7.5	7.4	5.5
1	5.2	7.1	8.5	9.8	11.1	12.2	13.0	13.4	13.2	9.8
1¼	8.1	11.1	13.2	15.4	17.4	19.0	20.4	20.9	20.6	15.3
1½	11.7	16.0	19.0	22.2	25.0	27.5	29.2	30.1	29.7	22.0
1¾	15.9	21.8	29.9	30.1	34.1	37.4	39.7	41.0	40.4	30.0
2	20.8	28.4	33.8	39.4	44.5	48.8	51.9	53.6	52.7	39.2

Gearing.

Where two shafts are so close together that neither belts nor chains can be employed, wheels whose peripheries are provided with teeth are used for transmitting power from one shaft to the other. This method is called transmission by gearing. Gears are

usually made of cast iron, though other materials are also used.

Kinds of Gearing.—According to the relative position of the axes of the gear wheels and the form of the teeth, six kinds of gearing may be distinguished: *spur-, bevel-, skew-, screw-, twisted-,* and *face-gearing*. The three first mentioned are most commonly used.

In *spur-, bevel-,* and *skew-gearing,* the surfaces of the teeth are formed by right lines; two engaging teeth of a pair of wheels touch each other along a right line and the teeth are thus merely rectilinear elements of the pitch surfaces.

The axes of spur wheels are parallel and the pitch surfaces are cylinders. The axes of bevel wheels intersect and the pitch surfaces are cones. The axes of skew wheels lie in different planes and the pitch surfaces are hyperboloids.

Figure 164. Gears.

The *twisted wheels* are merely a modified form of spur or of bevel wheels. In *screw gearing* the teeth are of helicoid form and may be regarded as helical elements of cylinders, which, since the axes are not in the same plane, are tangent to each other in one point only.

Face wheels have their axes perpendicular to each other and the teeth consist of turned pins. This gearing is not used any more in modern machinery.

Figure 164 illustrates three forms of gearing.

Pitch, Pitch Line and Pitch Diameter.—The diameter of a gear is the diameter of a circle which passes through the middle of all the teeth and which is called *pitch line*. Its diameter is designated by the name *pitch diameter*.

The distance from the center of one tooth to the center of the next, measured along the pitch line, is called the *circular pitch* of the gear.

The ratio between the number of teeth of a wheel and the diameter of the pitch circle is called the *diametral pitch.* For instance, if the pitch diameter of a wheel is 11 inches and it has

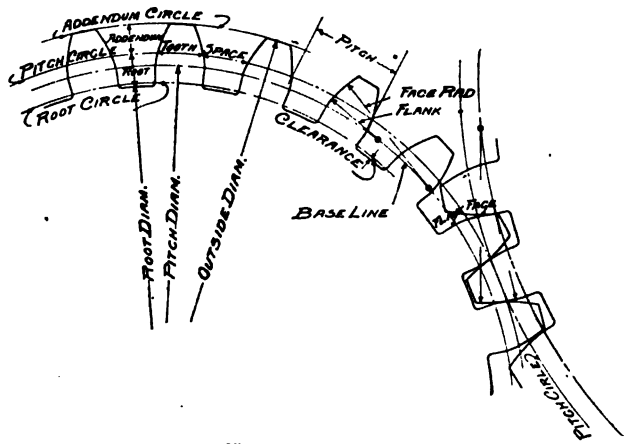


Figure 165. Parts of Gear Wheel.

44 teeth, then the diametral pitch is $\frac{44}{11}$ or 4. Such a wheel is said to have a 4 pitch. The circular pitch of the same wheel would be $\frac{11 \times 3.1416}{44}$ or 0.7854 inch.

Relation Between Diametral and Circular Pitch.

Diametral Pitch	12	10	8	7	6	5	4
Circular Pitch	0.2618 ¼	0.3142 ⅜	0.3927	0.4488 ½	0.5236 $\frac{9}{16}$ ⅝	0.6283 ¾	0.7854 ⅞ 1
Diametral Pitch	3	2¾	2½	2	1½	1	
Circular Pitch	1.0472 1⅛	1.1424 1¼	1.2566 1⅜ 1½	1.5708 1¾ 2	2.0944 2¼ 2⅜ 2¾ 3	3.1416 3¼ 3½	

When two wheels gear together, the circular pitch must be the same in each, and the ratio between the number of teeth of

both wheels must be the same as the ratio between the diameters of the pitch circles.

If the outer diameter is given (and this must be known, as the gear body must first be turned to it in the lathe before the teeth are cut), and if the number of teeth is given, the pitch diameter is obtained by first subtracting from the outside diameter 0.6 of the pitch.

By dividing the pitch circle, which can be calculated from this diameter, by the number of teeth, the circular pitch is obtained.

Dimensions of Teeth.—The number 0.6 is twice the height of the tooth above the pitch line, while 0.4 is the distance from the pitch line to the root of the tooth. The total height of the latter is therefore 0.7 of the pitch.

For large spur wheels the teeth should not be made so high, but preferably only 0.35 to 0.4 of the pitch, because they will then be more durable.

If it were possible to make the teeth with absolute accuracy, the breadth of the space between two teeth would be exactly equal to the breadth of each tooth; in reality, the spaces are always a little wider. The difference between the breadth of the tooth and of the space is called *back lash.*

In cast wheels the back lash is made a certain fraction, viz., $^1/_{15}$ to $^1/_{11}$ of the pitch, but in cut gearing it is made as small as the skill of the workman allows.

The width of the faces of the teeth is usually made from 2 to 3 times the circular pitch, but there is no fixed rule.

The outside line of the teeth is called the *addendum line,* and the root line the *dedendum line.* The depth to which a tooth enters the space between two teeth is called the *clearance line.*

Speed of Gearing.—The speed of rotation is inversely proportional to the number of teeth or to the diameter of the wheels. A wheel, for instance, which has 3 times as many teeth as another wheel, will revolve only ⅓ as rapidly. The method of calculation is the same as for pulleys, with the exception that the number of teeth is taken instead of the diameter of the wheels.

Example.—How many revolutions will a gear wheel which

has 24 teeth make, if driven by a smaller one with 12 teeth, making 90 revolutions per minute?

Solution.—$r = \frac{12 \times 90}{24} = 45.$

The large gear wheel will make 45 revolutions per minute.

The maximum pitch line speeds of different kinds of gearing are the following:

Ordinary cast iron wheels.....1,800 feet per minute
Helical cast iron wheels.....2,400 feet per minute
Mortice cast iron wheels.....2,400 feet per minute
Ordinary cast steel wheels.....2,600 feet per minute
Helical cast steel wheels.....3,000 feet per minute
Special cast iron machine-cut wheels.....3,000 feet per minute

Horse-power Transmitted.—The number of horse-power cast iron gearing can safely transmit may be calculated by means of the following formula:

$$\text{H. P.} = \frac{p \times w \times d \times r}{630}$$

p = circular pitch.
w = width of face of tooth in inches.
d = diameter of pitch circle in inches.
r = number of revolutions per minute.

Example.—How many H. P. can a wheel transmit which has a diameter of 16 inches, 3 pitch, a width of face of 4 inches, and makes 95 revolutions per minute?

Solution.—$\text{H. P.} = \frac{3 \times 4 \times 16 \times 95}{630} = 29.$

29 H. P. can be transmitted safely.

By transposing the above formula, the circular pitch, width of face, pitch diameter, and number of revolutions can be found, the transposed formulas being:

$$p = \frac{\text{H. P.} \times 630}{w \times d \times r}$$

$$w = \frac{\text{H. P.} \times 630}{p \times d \times r}$$

$$d = \frac{\text{H. P.} \times 630}{p \times w \times r}$$

$$r = \frac{\text{H. P.} \times 630}{p \times d \times w}$$

Example—A gear wheel is 16 inches in diameter, has 4 inches width of face, and is 3 pitch. How many revolutions per minute must it make in order to transmit 29 H. P.?

Solution.—$r = \frac{29 \times 630}{3 \times 16 \times 4} = 95.$

The gear wheel must make 95 revolutions per minute.

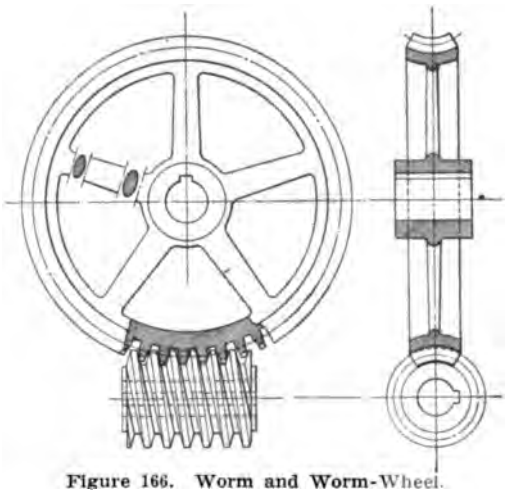


Figure 166. Worm and Worm-Wheel.

Worm Gear (Screw Gear).—This gear is very frequently applied for very large transmission ratios—1 to 100 and higher. The worm is the driver, and the device is self-locking, that is, no back motion can be started from the gear side. Figure 166 illustrates a worm gear.

Pumps

A pump is an apparatus for raising water or other liquids from a lower level to a higher level.

This lifting of water can be done either with the help of the atmospheric pressure by means of *lift or suction pumps,* generally designated as *piston pumps,* or by *force pumps,* in which some force other than the pressure of the atmosphere is utilized.

Piston Pumps.

A piston or suction pump does not "lift" water, properly speaking, but only removes the air at one point of the water surface. The air pressure resting on the remaining water surface then forces the water through the suction pipe into the vacuum formed under the ascending piston.

Height to which Water can be Lifted.

Theoretically a piston pump could lift water at sea level almost 34 feet, or, more accurately, 33.95 feet, the height corresponding to a perfect vacuum. It is impossible, however, to create such a vacuum over a water surface, as water always contains free air and other gases, which would escape into the empty space, and, furthermore, the water itself would vaporize, thus destroying the vacuum. For these reasons water cannot be lifted more than approximately 25 feet high by means of suction pumps. The height to which water can be pumped decreases as the temperature of the water increases, and when the water reaches boiling temperature, it cannot be lifted by suction, but must flow into the pump by gravity. Many expensive installations have been made and have proven failures, for the reason that these facts were ignored, although they have long been known.

Calculation of Height of Water.—The height to which a pump can theoretically lift water is easily calculated, if the tem-

perature of the water is known. Let us assume, for instance, that the water has a temperature of 202° F.

The absolute pressure of steam having the same temperature as the water, viz., 202° F., is 12 pounds per square inch, and the pump trying to lift the water of 202° F. will not be able to create a vacuum any higher than 12 pounds. Consequently, the pressure available for lifting the water amounts to 14.7—12 or 2.7 pounds per square inch. Since a pressure of 0.434 of a pound at sea level is able to lift water one foot high, the total height to which water of 202° F. can theoretically be pumped is $\frac{2.7}{0.434}$, or 6.2 feet.

Duty of Pumps.

By the term *duty of a pump* is understood the amount of work in foot-pounds performed by a pump when furnished with 1,000,000 B. T. U. from the boiler. Formerly the calculation of the duty of a pump was based on 100 pounds of coal used in furnishing the necessary energy, but in 1891 the more accurate basis of 1,000,000 B. T. U. was adopted by the American Society of Mechanical Engineers.

The formula for the duty of a pump is:

$$\text{Duty} = \frac{\text{Foot-pounds of work obtained} \times 1{,}000{,}000}{\text{Total number of B. T. U. expended.}}$$

Theoretical Capacity of Piston Pumps.

The formula for calculating the theoretical discharge in gallons per minute is:

$$g = \frac{a \times s \times n}{231}.$$

g = gallons of water delivered per minute.
a = area of water cylinder in square inches.
s = length of stroke in inches.
n = number of strokes per minute.
231 = cubic inches contained in one gallon.

Example.—A pump has a 6-inch water piston, a 10-inch stroke, and makes 120 strokes per minute. How many gallons does it theoretically discharge per minute?

Solution.—$g = \frac{6 \times 6 \times 0.7854 \times 10 \times 120}{231} = 146.88.$

The theoretical discharge of the pump is 147 gallons per minute.

A good rule, and one easily remembered, is to multiply the piston speed in feet per minute by $4 \times d^2$ and divide by 100. This gives very closely the theoretical discharge in gallons per minute. The piston speed in feet per minute is the length of stroke in feet multiplied by the number of strokes per minute.

Practical Capacity of Piston Pumps.

There is always a certain loss of water or *slippage* through valves and piston, and the quantity of water delivered by the pump is reduced correspondingly. The actual capacity of a pump varies from 60 per cent to 95 per cent of the theoretical, according to design, workmanship, discharge pressure, etc., and on the average it amounts to approximately 80 per cent.

Based on an actual capacity of 80 per cent of the theoretical, the following formula gives the practical capacity of a pump in gallons per minute:

$$g = 0.00272 \times d^2 \times l \times n.$$

g = gallons of water delivered per minute.

d = diameter of water-cylinder in inches.

l = length of stroke in inches.

n = number of strokes per minute.

This formula is derived as follows:

Cubic inches discharged per minute $= 0.80 \times 0.7854 \times d^2 \times l \times n$.

One gallon = 231 cubic inches.

Hence, gallons per minute $= \dfrac{0.80 \times 0.7854 \times d^2 \times l \times n}{231}$

But, $\dfrac{0.80 \times 0.7854}{231} = 0.00272.$

Hence, $g = 0.00272 \times d^2 \times l \times n$.

Example.—Assuming the actual capacity to be 80% of the theoretical, how many gallons per minute will be discharged by a pump having a water-cylinder 5 inches in diameter, a 10-inch stroke and making 100 strokes per minute?

Solution. — $g = 0.00272 \times 5 \times 5 \times 10 \times 100 = 68$ gallons per minute.

Size of Suction and Discharge Pipes.

The diameter of the suction or the discharge pipe of a pump for a given flow per minute can be found as follows:

$$d = 4.95 \times \sqrt{\frac{\text{gallons per minute}}{\text{velocity in feet per minute}}}$$

The derivation of this formula is as follows:

Cubic inches per minute = velocity in inches per minute $\times$ 0.7854 $\times$ d^2.

$$\text{Diameter} = \sqrt{\frac{\text{cubic inches per minute}}{\text{velocity in inches per minute} \times 0.7854}}$$

$$= \sqrt{\frac{\text{gallons per minute} \times 231}{\text{velocity in feet per minute} \times 12 \times 0.7854}}$$

$$\text{But, } \sqrt{\frac{231}{12 \times 0.7854}} = \sqrt{24.51} = 4.95$$

$$\text{Hence, } d = 4.95 \sqrt{\frac{\text{gallons per minute}}{\text{velocity in feet per minute}}}$$

Example.—What must be the size of the pipe of a pump which is to deliver 150 gallons per minute at a velocity of 200 feet per minute?

Solution.—$d = 4.95 \times \sqrt{\frac{150}{200}} = 4.28$ inches.

The nearest size obtainable would be a 4-inch pipe.

Horse-Power Required for Piston Pumps.

The theoretical horse-power required to pump a certain quantity of water to a certain height can be found by multiplying the number of pounds of water per minute by the height in feet to which it is to be lifted, and dividing the product by 33,000. If a certain pressure in pounds per square inch is given, against which the water is to be pumped, we can find the corresponding height of water by multiplying by 2.31, and then calculate the horse-power as before. The theoretical horse-power required to raise water to a certain height may be expressed by the following formula:

$$\text{H. P.} = \frac{W \times H}{33{,}000}$$

H. P. = horse-power required.

W = pounds of water lifted per minute.

H = height in feet to which water is lifted.

For large and well-constructed pumps 25 per cent should be added to the result for friction and leakage, while in smaller pumps of inferior construction about 50 per cent must be added.

Example.—How many horse-power are theoretically necessary to pump 144 gallons of water (1 gallon = 8.34 pounds) per minute 175 feet high?

Solution.— $\frac{144 \times 8.34 \times 175}{33{,}000} = 6.37$ horse-power.

The actual horse-power could then be found by adding from 25 to 50 per cent to this result. Thus, if 50 per cent were to be added, we would have 6.37 + 50 per cent of 6.37 = 6.37 + 3.19 = 9.56 horse-power.

Cost of Operation.

If the above pump uses 120 pounds of steam per horse-power per hour, and if it costs 18 cents for fuel to produce 1,000 pounds of steam, the cost of operation of the pump under the above conditions would amount to $\frac{9.56 \times 120 \times 18}{1{,}000} = 20.6$ cents per hour.

Conditions for Satisfactory Operation of Piston Pumps.

For satisfactory operation of piston pumps, the following important conditions should be fulfilled. The water supply should be full and steady. The suction and discharge pipes should in no case be smaller than the openings in the pump, and the suction lift and delivery pipe should be as straight as possible and smooth inside. An air chamber should be placed on the discharge side of the pump to prevent pounding and to secure a steadier outflow. Sometimes such an air chamber is placed in the suction side to make the action of the pump easy and uniform, and this also facilitates the filling of the barrel of the pump at high speed. Long suction pipes should be provided with a foot valve just above the strainer in the well or pit.

The piston speed is of considerable importance. For pumps of usual construction and of medium size, having a water cylinder with a diameter of from 4 to 10 inches, the economical and average speed for continuous service is between 25 and 40 feet

of piston speed per minute, for small sizes 30 to 65 feet, for large sizes 60 to 100 feet per minute.

The velocity of the water passing the valves should not exceed 300 feet per minute, while the velocity of the water in the suction pipe should not exceed 200 feet per minute; otherwise the frictional resistance to the suction becomes too great, and the cylinder may only become partly filled and the capacity of the pump reduced correspondingly. The velocity of flow of water in the discharge pipe should not exceed 300 feet per minute. Generally the speed of the water in both the suction and discharge pipes is between 150 and 300 feet per minute. Since friction increases rapidly with the velocity, viz., as the square of the latter, it is preferable to keep the velocity of the water at from 150 to 300 feet per minute. The author frequently found the discharge pipe to be unreasonably reduced in size, resulting in an enormous friction caused by the water passing through the pipe at a high velocity.

Types of Piston Pumps.

Piston pumps have either a piston or a plunger. A *piston* moves in a cylinder, in which it fits tight. A *plunger* fits tight in the stuffing box only, while the cylinder has a diameter somewhat greater than the plunger diameter. They may also be designated as *direct-acting steam pumps,* and as *power-pumps,* according as they are equipped with a steam cylinder and have steam and water piston on the same piston rod, or are driven from an engine, a windmill, a shaft, etc.

Direct-Acting Steam Pumps.

In this country, the direct-acting steam pump is most commonly used. It is either made single-acting, double-acting, or duplex.

The *single-acting pump* sucks and discharges only once during each complete piston motion.

The *double-acting* pump sucks on one side while it discharges at the same time on the other.

The *duplex* pump consists of two double-acting pumps joined side by side. The steam is regulated by one pump for the other.

The reason that direct-acting steam pumps have a wider field

of distribution than pumps of other constructions is that they are cheap, strong, durable, easily and quickly installed, and operated without difficulty. Figures 167 to 169 illustrate the construction of various types of direct-acting steam pumps.

Economy.—Direct-acting pumps are very low in economy. The amount of heat utilized by them is often less than 2% of the heat value in the coal, 98% being lost. Conditions are nat-

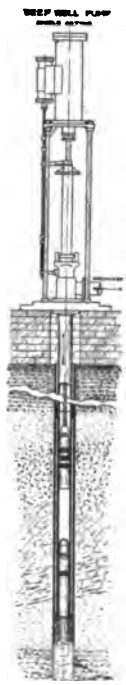


Figure 167. Deep-Well Pump.

urally more favorable if the exhaust steam is utilized for various purposes. A simple calculation will show the wastefulness of these pumps. If we assume, for instance, that a direct-acting steam pump uses 120 pounds of steam per horse-power per hour, and that the boiler evaporates 8 pounds of water per pound of coal, there are then 15 pounds of coal used per horse-power per hour. The number of foot-pounds generated in the furnace per minute, assuming the fuel value of each pound of coal to be

12,000 B. T. U., would be $\frac{12{,}000 \times 15 \times 778}{60} = 2{,}334{,}000$, and as 33,000 foot-pounds per minute are equal to one horse-power, the percentage of efficiency would be $\frac{33{,}000 \times 100}{2{,}334{,}000} = 1.41$ per cent. This illustrates sufficiently the low economy of direct-acting steam pumps. The steam acts like a solid plug in them and no expansion takes place. However, these pumps should be bought for simplicity and convenience, and not for economical operation.

How Pumps are Designated.—The dimensions of a pump are always given in the same order:

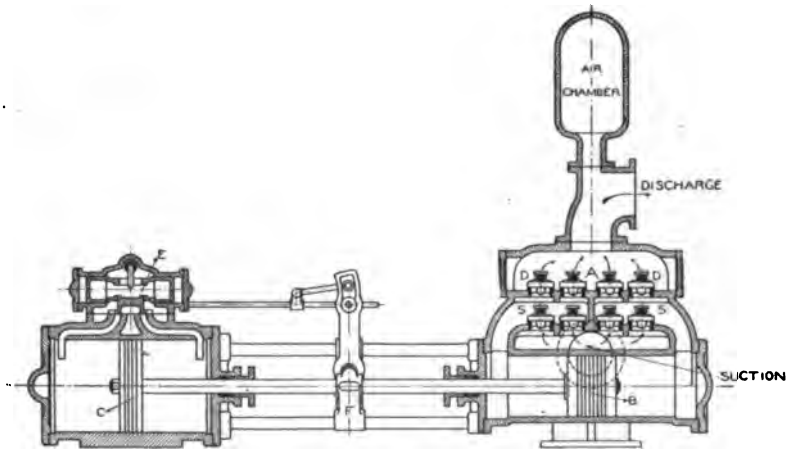


Figure 168. Double-Acting Pump.

1. Diameter of steam cylinder in inches.
2. Diameter of water cylinder in inches.
3. Length of stroke in inches.

Thus, a $6 \times 4 \times 6$ pump is one in which the diameter of the steam cylinder is 6 inches, diameter of water cylinder 4 inches, and length of stroke 6 inches.

Duty of Direct-Acting Steam Pumps.—The average duty of direct-acting steam pumps may be taken to be 15,000,000 foot-pounds, obtained from 1,000,000 B. T. U. furnished by the boiler. In large plants equipped with high-grade pumping engines, this duty is exceeded 10 times.

Size of Steam and Water Cylinders.—The areas of the steam and the water cylinders of a direct-acting steam pump are inversely proportional to the pressure.

The diameter of the steam cylinder can be found as follows:

$$d_1 = \sqrt{\frac{P \times d^2}{P_1}}$$

The diameter of the water cylinder is:

$$d = \sqrt{\frac{P_1 \times d_1^2}{P}}$$

d_1 = diameter of steam cylinder in inches.
d = diameter of water cylinder in inches.

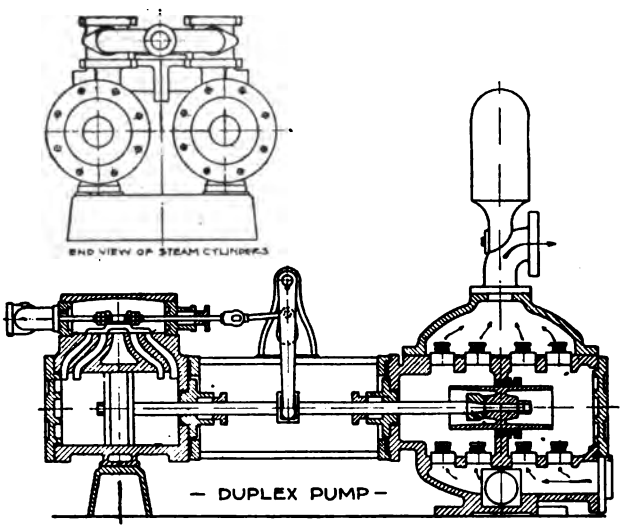


Figure 169. Duplex Pump.

P = pressure in water cylinder in pounds per square inch.
P_1 = pressure in steam cylinder in pounds per square inch.

Example.—What must be the diameter of the steam cylinder, if the steam pressure is 36 pounds per square inch, the water pressure 100 pounds per square inch, and the diameter of the water cylinder 4.5 inches?

Solution.—$d_1 = \sqrt{\frac{100 \times 4.5 \times 4.5}{36}} = 7.5.$

The steam cylinder must be 7.5 inches in diameter.

Calculation of Steam Pressure and Water Pressure.—The steam pressure necessary for a pump of given steam and water cylinder diameters and given water pressure can be found thus:

$$P_1 = \frac{P \times d^2}{d_1^2}$$

The water pressure can be found as follows:

$$P = \frac{P_1 \times d_1^2}{d^2}$$

The explanation of the letters in these two formulas is given on the previous page.

Example.—What steam pressure must be used, if the diameter of the steam cylinder is 6 inches, diameter of water cylinder 4 inches, and the water pressure 90 pounds per square inch?

Solution.—$P_1 = \frac{90 \times 4 \times 4}{6 \times 6} = 40.$

The steam pressure must be 40 pounds.

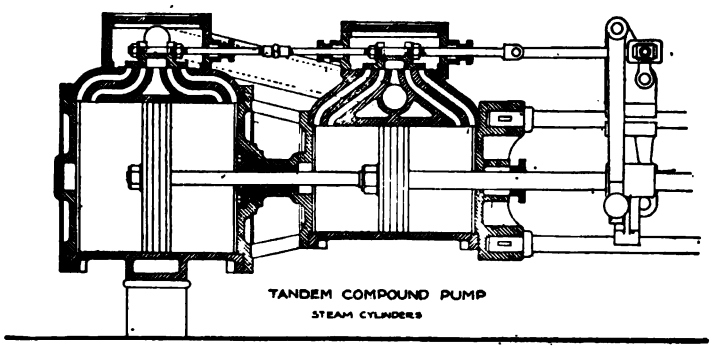


Figure 170. Tandem Compound Pump.

Size of Boiler Feed Pumps.—In selecting boiler feed pumps, which are always direct-acting, one should specify the pump to be capable of pumping 7 pounds of water for every square foot of boiler heating surface per hour, at 40 feet piston speed. This would give a pumping capacity of 42 gallons per minute for 3,000 square feet of boiler heating surface. The area of the steam cylinder should be 2¼ to 2¾ times larger than that of the water cylinder.

Compound Pumps.

Where large quantities of water have to be pumped, so-called compound pumps, in which the steam is utilized to a considerably larger extent, are used. Sometimes these compound pumps are made condensing.

Construction.—Compound pumps are equipped with two steam cylinders of different sizes, usually arranged in tandem, as is illustrated by Figure 170.

If the pumps are non-condensing, two expansions of the steam are obtained, but if made condensing, about four expansions take place.

Dimensions and Capacities of Direct-Acting Steam Pumps.

Diameter of Steam Cylinder in Inches.	Diameter of Water Plunger in Inches.	Length of Stroke in Inches.	Diameter of Plunger and Length of Stroke required in Single Cylinder Steam Pump to do the same work.		Gallons delivered per minute at a speed that can be taken as a basis of comparison.	
			Plunger, Inches.	Stroke, Inches.	Piston Speed, Feet Per Minute.	Gallons.
2	1⅛	2¾	1⅝	5	50	5
3	1¾	3	2½	6	50	12
3	2	3	2⅞	6	50	15
4½	2¾	4	4	7	50	30
4½	3¼	4	5¼	7	50	55
5¼	3½	5	5	10	50	50
5¼	4¾	5	6¾	10	50	90
6	4	6	5⅝	10	50	65
7½	5	6	7	10	50	100
6	5¾	6	8⅜	10	50	135
7½	4½	10	6⅜	16	100	165
9	5¼	10	7½	16	100	225
	6	10	8½	16	100	295
	7	10	9⅞	16	100	400
	8½	10	12	16	100	590
	9¼	10	13	16	100	700
	10¼	10	14¼	16	100	855
	12	10	17	16	100	1175
	14	10	19¾	16	100	1600
	10	15	14	24	100	815
	11	15	15½	24	100	990
	12	15	17	24	100	1175
	15	15	21	24	100	1835
Varying with kind of work and steam pressure.	16	18	22⅝	36	100	2085

How to Set Duplex Pumps.—Duplex pumps are often packed unequally, one piston being packed more tightly than the other. More frequently, the slide valves are not set right, with

Diameters of High and Low Pressure Cylinders of Compound Pumps Corresponding to Different Ratios of Piston Areas.

Diameter High Pressure Cylinders.	Ratio of Piston Areas.				
	2 to 1	2½ to 1	3 to 1	3½ to 1	4 to 1
	Diameter of Low Pressure Cylinders.				
6	8½	9½	10½	11	12
7	10	11	12	13	14
8	11	12½	14	15	16
9	13	14	15½	17	18
10	14	16	17	19	20
11	15½	17½	19	20½	22
12	17	19	21	22½	24
13	18½	20½	22½	24½	26
14	20	22	24	26	28
15	21	24	26	28	30
16	22½	25	28	30	32
17	24	27	29½	32	34
18	25½	28½	31	33½	36
19	27	30	33	35½	38
20	28	31½	34½	37½	40
21	30	33	36½	39	42
22	31	35	38	41	44
23	32½	36½	40	43	46
24	34	38	41½	45	48

the result that the pump works one-sided. The proper method for setting a duplex pump is as follows:

Place the steam pistons in the center of their travel, which will bring the rocker arms perpendicular to the rods. Place the slide-valve over the center of the steam ports. The lost motion on the valve stems in pumps of small size that usually have non-adjustable blocks, should measure the same on each side of the blocks between the valve lugs. If the lost motion is equal, the valves are set. Move one valve so as to admit steam before putting on the steam chest covers. On the medium and larger sizes, where adjustable lock nuts are used on the valve stems, there are no fixed rules for the exact amount of lost motion, which governs the length of stroke, and this can only be determined satisfactorily by trial and careful adjustment. In the large pumps with the outside slotted link and sliding block, where the valve movement is coincident in time and amount with the valve rod movement, the distance from each end of the link block to each side of the link slot should measure the same when the slide-

valves are central over the ports. The valves are then properly set and the pump will work smoothly.

Power Pumps.

A power pump, i. e., a pump run by belt or electric motor, is the type to be used if economy is the object. If different speeds are required, the belt-driven power pumps should have a step-cone pulley, so that three different speeds can be obtained, viz., a minimum, a medium, and a maximum speed.

If electricity is the driving agent the speed of the pump can be reduced or increased at will if a speed controller is attached to the motor.

Power pumps may be either *single, double* or *triplex,* according to the number of water cylinders the pump is equipped with.

Economy.—If the power is supplied by an automatic cut-off engine the economy of a power-driven pump is very much higher than that of a direct-acting steam pump. An engine of the Corliss type uses about 30 pounds of steam per horse-power per hour, as against 120 pounds of steam used by a direct-acting steam pump for the same amount of work.

FORCE PUMPS.

While in the piston pumps, as has been remarked before, the water is lifted by atmospheric pressure, force pumps utilize other powers for this purpose, for instance, the energy of steam or electricity, or the momentum of water in motion.

We distinguish between *centrifugal* pumps, *rotary* pumps, *pulsometer* and *hydraulic ram.*

Centrifugal Pumps.

Where large quantities of water are to be lifted to a moderate height, centrifugal pumps, driven by belt, electric motor, steam engine or turbine, are often used. Figure 171 shows such a pump.

Construction and Operation.—In this pump, the water enters a hollow-armed or a propeller-shaped wheel, called the impeller, near its center. The wheel rotates with great velocity, and, by the centrifugal force developed by the rotatory motion, throws the water towards the casing, producing a high pressure, which forces a continuous stream of water through the discharge pipe.

In these pumps it is essential that all sudden changes of

direction in the path of the water through the pump be avoided. In some pumps the bends of the arms should not have too small a radius; the vanes should have smooth surfaces and be shaped so as to receive the water without shock. The water space must increase toward the discharge opening so that there is no change in velocity inside the casing. The casing receives thereby a spiral-shaped form. The maximum velocity of the water in relation to the vanes should not exceed 450 feet per minute.

Efficiency.—Centrifugal pumps are most economical for large quantities of water, and if the height to which the water is

Figure 171. Centrifugal Pump (Taber).

raised does not exceed 65 feet; for higher lifts the power consumption grows rapidly. For higher heads than 65 feet, it is necessary to surround the moving wheel with a concentrical guiding apparatus called the *diffuser,* whereby the high velocity of the water is transformed into pressure. By this means, a head as high as 120 feet is obtained from one wheel only.

Since there is a steady stream without need of a suction valve, the suction lift is more than with a piston pump, being at a maximum 27 feet. On the other hand, the *efficiency* of the centrifugal pump is *lower* than that of the piston pumps. This disadvantage, however, is more than counterbalanced by the small space required, the little attention which is necessary, and the

possibility to drive it directly with an electric motor or a steam turbine on the same shaft. Besides these there are a few other advantages as concerns the quantity regulation, the absence of valves and air chambers and the possibility of handling viscid and muddy liquids. It is necessary, however, that the pump be filled with water to obtain a suction when starting up, and in case the pump is installed above the lower level, a valve must be installed, which is to be closed at standstill so as to keep the water inside the pump.

By changing the speed of the pump, the quantity delivered may be varied through a very wide range, or, if this cannot be done, a valve may be arranged in the pressure pipe, whereby the amount of water can be regulated with little influence upon the efficiency. When this valve is completely closed, the pump therefore rotating idly, the power consumed amounts at a maximum to 40% of the nominal rating.

The *height of lift* depends on the tangential velocity of the circumference; constant circumferential speed gives a constant head independent of the size of the pump. The *quantity of water* discharged is in direct proportion to the area of the discharging orifices at the circumference.

The efficiency of these pumps seems to be greatest at a lift of 17 feet, and experience shows it to amount to about 60%; however, it increases with the size of the pump. Recent designs of centrifugal pumps have shown as high as 80% efficiency.

Two-Stage and Multiple-Stage Centrifugal Pumps.—For high lifts, two or more stages are used, i. e., two or more centrifugal pumps are so arranged and connected, that the water is taken through the pumps consecutively. Figure 172 shows a so-called multiple centrifugal pump, used for very high lifts.

Centrifugal pumps of this type are able to raise the water from 100 to 2,000 feet, and it is claimed that they maintain an efficiency of from 70 to 80 per cent.

Size of Suction and Discharge Pipes.—The diameter of the suction or of the discharge pipe of a centrifugal pump can be found by any of the following formulas:

$$1.\quad d = 0.081 \times \sqrt{W}.$$

$$2.\quad d = 0.255 \times \sqrt{G.}$$

$$3.\quad d = 0.638 \times \sqrt{C.}$$

d = diameter of discharge or suction pipe in inches.
W = pounds of water per minute.
G = gallons of water per minute.
C = cubic feet of water per minute.

Example.—What must be the diameter of the suction and discharge pipes of a centrifugal pump, if 400 gallons are to be delivered per minute?

Solution.—According to formula 2, we have $d = 0.255 \times \sqrt{400} = 5.1$ inches. A 5-inch pipe must therefore be used.

Figure 172. Multistage Centrifugal Pump.

Number of Revolutions of Impeller.—To find the number of revolutions necessary for an impeller of given diameter and for a certain height of delivery, the following formula can be used:

$$N = k \times \frac{\sqrt{H}}{D}$$

N = the number of revolutions per minute.
D = diameter of wheel in feet.
H = height of delivery in feet.
k = 153 to 248 according to size of pump, and may be taken as 153 for small, and 187 for large pumps.

Example.—How many revolutions must the impeller of a

centrifugal pump make to deliver 400 gallons of water per minute to a height of 65 feet, if the diameter of the impeller is 2 feet?

Solution.—$N = 153 \times \frac{\sqrt{65}}{2} = 617.$

The impeller must make 617 revolutions per minute.

Turbine Pumps.—These pumps are generally designed to be directly connected to a steam turbine by a flexible coupling, and the best results are obtained at a high peripheral velocity.

Rotary Pumps.

The underlying principle of the rotary pump can easily be seen from Figures 173 and 174. After the water has entered the housing it is gradually forced into the discharge pipe, by the

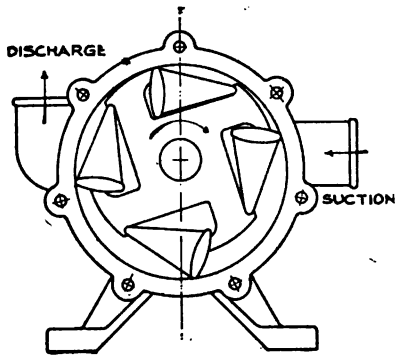


Figure 173. Rotary Pump (Section).

rotation of interlocking wheels or of movable wings rotating eccentrically, which gradually narrow the space and press the water into the discharge pipe.

Operation.—These pumps should be run slowly, for they are very noisy and wear out rapidly, if run at excessive speed. Generally they are operated at from 50 to 150 revolutions per minute, depending somewhat on the type. They are more suitable to pump against high pressure than centrifugal pumps, as they consume considerably less power in such cases. They are not adapted for gritty water or liquids carrying rough impurities.

Calculation of Number of Revolutions.—The velocity of the water in the discharge pipe should be from 150 to 200 feet per minute, and from this can be calculated the number of revo-

lutions the wheel has to make, by simply dividing the velocity by the inside circumference of the housing.

Figure 174. Rotary Pump (Taber).

Example.—If the housing of a rotary pump is one foot inside diameter, how many revolutions must it make if the velocity of the water is to be 180 feet per minute?

Solution.—The inside circumference of the housing is 1×3.1416, or 3.1416 feet, and the number of revolutions is therefore:

$$180 \div 3.1416 \text{ or } 57 \text{ per minute.}$$

Pulsometer.

The pulsometer, as will be observed from Figure 175, consists of two bottle-shaped chambers joined side by side, with tapering necks bent toward each other and uniting in one common upright passage. Into this a small ball is fitted, so as to oscillate with a slight rolling motion between seats formed in the junction, which are alternately opened and closed for the admission of steam. Each chamber has a separate suction and discharge valve. The air chamber is cast between the two bottle-shaped chambers. A small brass check valve is screwed into the air chamber, and one into the vacuum chamber. The check valve in the neck of each chamber allows a small quantity of air to enter above the water, to prevent the steam from agitating it on first entrance, and to diminish the condensation of steam during the discharge

stroke. The check valve in the vacuum chamber serves to cushion the blow of the water consequent upon the filling of each chamber alternately.

Operation of Pulsometer.—When all chambers and pipes are empty, the air check valves should be closed and the globe valves opened for an instant. Steam will enter one of the chambers, expel the air and condense, forming a vacuum. This operation being repeated several times, both chambers will be filled

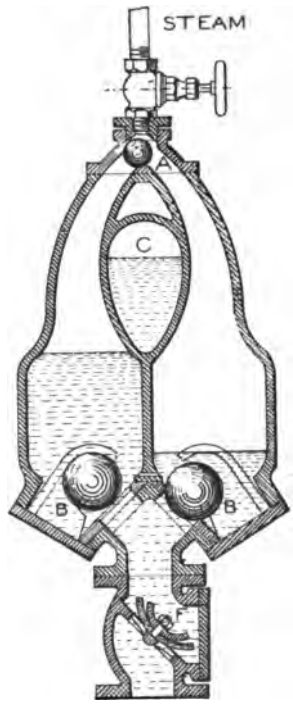


Figure 175. Pulsometer.

with water through the inlet pipe. The air valves in the chambers must now be opened a little to secure a regular and continuous action, noticeable by a steady pulsation and smooth working of the steam ball without a rattle. Steam, being now permanently admitted, enters the chamber not closed by the ball, and forces the water through the discharge valves until the surface of the water is lowered below the discharge orifice. At that instant

the steam begins to escape into the discharge pipe and to condense, forming a partial vacuum in the chamber. The water in the other chamber now presses the ball, which rolls over and closes the first chamber, where the water enters through the inlet valve to fill the vacuum. This operation alternately changes from one chamber to the other. One chamber fills with water in about two-thirds of the time required for the other to empty. The expenditure of steam is proportional to the delivery, and independent of the elevation; the pulsations and delivery vary in inverse proportion to the lift.

The velocity of the water discharged by the pulsometer should not be too high. Generally the velocity amounts to approximately 240 feet per minute.

Duty of Pulsometer.—The duty of a pulsometer ranges from 12,000,000 to 15,000,000 foot-pounds per 1,000,000 B. T. U. It can be calculated according to the following formula:

$$D = \frac{H \times 1{,}000{,}000 \times 100}{(t_1 - t) \times (100 - F)}$$

D = duty in foot-pounds per 1,000,000 B. T. U. furnished by boiler.

H = height in feet to which water is lifted.

F = friction of water in per cent.

t_1 = final temperature of water in degrees F.

t = initial temperature of water in degrees F.

The *efficiency* of the pulsometer in per cent can be found as follows:

$$E = \frac{H \times (100 + F)}{(t_1 - t) \times 778}$$

Example.—A pulsometer lifts 175 gallons of water per minute to a height of 38 feet (including suction and delivery lift) with steam at 50 pounds gauge pressure. The initial temperature of the water is 60° F., the final temperature is 63.5° F. The friction amounts to 10%. What is the duty in foot-pounds and the efficiency in per cent?

Solution.—$D = \frac{38 \times 1{,}000{,}000 \times 100}{(63.5 - 60) \times (100 - 10)} = 12{,}063{,}492.$

The duty amounts to 12,063,492 foot-pounds.

$$E = \frac{38 \times (100 + 10)}{(63.5 - 60) \times 778} = 1.54.$$

The efficiency is 1.54%, or of 100 B. T. U. furnished by the steam, only 1.54 perform useful work.

It will be noticed in the above example that duty and efficiency can be calculated without knowing the quantity of steam used, or considering the steam pressure and quantity of water delivered, as long as the lift and the initial and final temperatures of the water are known.

Calculation of Diameter of Water-Pipe for Pulsometer.—The diameter of the water-pipe necessary for discharging a certain quantity of water can be found according to the following formula:

$$d = \sqrt{\frac{Q \times 231}{v \times k \times 0.7854 \times 12}}$$

d = diameter of pipe in inches.

Q = gallons of water delivered per minute.

v = velocity of water in feet per minute.

k = coefficient of discharge, usually assumed as 0.8.

Example.—What must be the diameter of the discharge pipe of a pulsometer, if 175 gallons of water are to be delivered per minute, with a velocity of 265 feet?

Solution.—

$$d = \sqrt{\frac{175 \times 231}{265 \times 0.8 \times 0.7854 \times 12}} = 4.5 \text{ inches.}$$

A 4½-inch pipe must be used.

Hydraulic Ram.

While in the devices described on the preceding pages, steam or some other power is employed for lifting water, the hydraulic ram utilizes the momentum of moving water. The construction, as will be perceived from the illustration, is extremely simple.

A quantity of water from a pond or reservoir is set in motion through an inclined tube, and its escape from a lower orifice is made suddenly to cease, when the momentum of the moving mass drives a portion of its own volume to an elevation much higher than that from which it descended.

Efficiency of Hydraulic Ram.—The efficiency of the ram is very low, as only about one-seventh of the water used is elevated, if the elevation is ten times the fall, and it can only be

employed where there is an ample supply of water and moderate fall.

Morin, from his own experiments and those of Eytelwein, gives the formula for efficiency as

$$0.258 \times \sqrt{12.8 - \frac{h}{H}}$$

h = height to which water is lifted.

H = head of water under which the ram works.

Example.—What is the efficiency of a ram working with a fall of 9 feet and raising the water 63 feet?

Solution.— $0.258 \times \sqrt{12.8 - \frac{63}{9}} = 0.62.$

The efficiency is 62%.

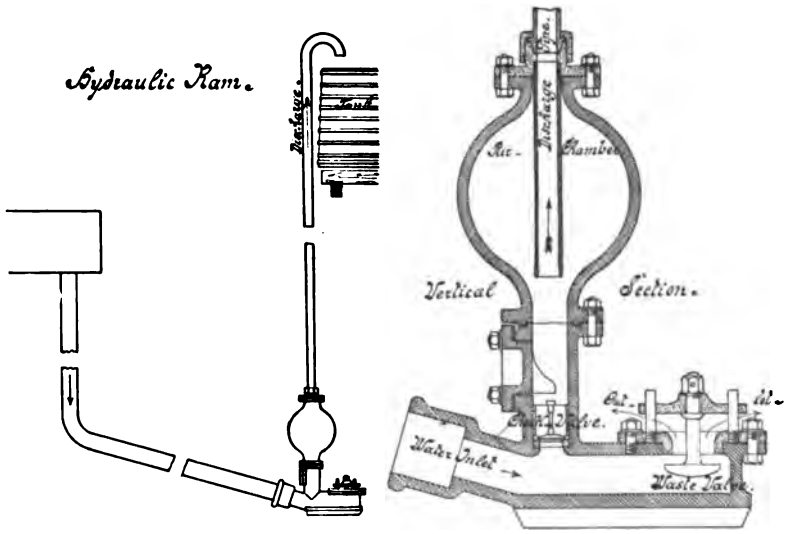


Figure 176. Hydraulic Ram.

Air Compressors

There are two common types of air compressors, the *straight line* air compressor and the *crank-driven* compressor. The *straight line* air compressor has a steam cylinder in tandem with the air cylinder, and both pistons are on one piston rod, with a heavy cross-head between, to which are coupled two connecting rods and two cranks on one crankshaft with two heavy fly-wheels. The latter must carry the full load over the dead center, for it is evident that the resistance against the air piston is the greatest when the steam pressure behind the steam piston is the weakest. A constant jerking on the bearings takes place in these compressors. Figure 177 shows its construction.

In the *crank-driven* air compressor, illustrated by Figure 178, the air cylinder is on one side and the steam cylinder on the other side, both connected to one crankshaft, with the cranks connected to each other at an angle of 90°.

This kind of compressor is generally preferable, because of greater durability and economy; it is however more expensive.

The power-driven compressor consists of one or more air cylinders, either single or double-acting, driven by a belt or electric motor.

Centrifugal air compressors are on the market, having a capacity of from 750 to 28,000 cubic feet of free air per minute.

Capacity and Economy.

The capacity of an air compressor is expressed by the amount of free air it can deliver per minute with a given number of strokes. By its economy is understood the amount of free air it can deliver against a certain pressure with the smallest amount of power expended.

Friction.

The friction in a steam-driven air compressor is about 10%

of the total horse-power required, whereas in a belt-driven compressor, it is frequently less than 5%.

Horse-power Necessary for Operation.

Practical tests have shown that about 0.2 horse-power is required to compress one cubic foot of free air to a gauge pressure of 100 pounds, about 0.189 H. P. for compressing to 80 pounds pressure; and about 0.159 H. P. for compressing to 60 pounds pressure. If it is desired, for instance, to have a compressor deliver 125 cubic feet of free air and to compress it to 100 pounds pressure, 25 H. P. would be required to do this; for 80 pounds pressure 23.6 H. P., and for 60 pounds pressure about 20 H. P.

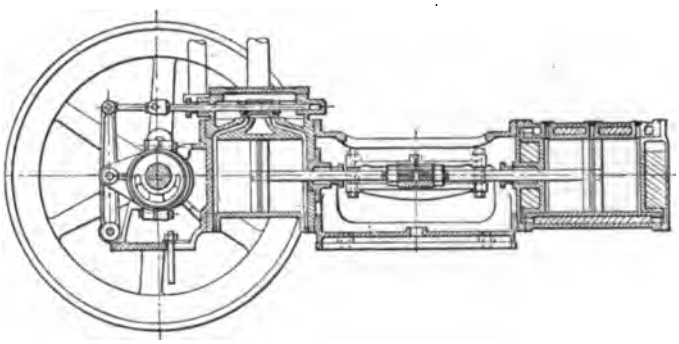


Figure 177. Straight Line Air Compressor

Loss of Power in Transmission.

When transmitting power from an air compressor to a motor which is at a great distance, for instance, a mile, from the source of power, about 10% of the power is lost.

Efficiency of Air-driven Motor.

Assuming the efficiency of the steam engine when running the air compressor direct connected to be 90%, the efficiency in compressing 75%, loss by transmission of power over a distance of one mile, 10%, and the efficiency of the motor, 50%, then 100 engine horse-power will furnish only $100 \times 0.90 \times 0.75 \times 0.90 \times 0.50 = 30.38$ horse-power delivered by the motor, or the efficiency is 30.38%.

Proper Method of Compressing Air.

The most economical way to compress air is in stages, i. e. to use two or more cylinders. In the first cylinder, the air is compressed to a low pressure; it then passes through a cooler which cools it to nearly the temperature of the cooling water, and from here the air enters the second cylinder, where it is further compressed. This process of compressing and cooling is continued until the desired pressure is obtained.

Two-stage or compound compressors are used for pressures between 60 and 300 pounds, three-stage or triple compressors for pressures ranging from 300 to 1,000 pounds, etc. To compress 100 cubic feet of free air to 75 pounds pressure requires about

Figure 178. Crank-Driven Air Compressor.

16 H. P. for a single cylinder and 13.75 H. P. with a compound cylinder, but if the air is compressed to 500 pounds pressure, 40 H. P. would be required with a single, and 30 H. P. with a compound cylinder, from which it can be seen that by using a compound cylinder a saving of 14% is effected for 75 pounds pressure, and of 25% for 500 pounds pressure. It is thus evident that compounding is very economical for high pressures.

The colder the air is to start with, the better, and the supply should therefore always be drawn from as cold a place as possible, since for every 5 degrees the temperature of the intaken

air is lowered, a saving of about 1% is effected in running the machine. The capacity is also increased. If in a brewery, for instance, the air is taken from the cellars at a temperature of 35° F., instead of from the engine room, where the temperature will often during the hot season reach 85° F., a saving of (85 — 35) ÷ 5, or 10% is effected.

Final Temperature of Compressed Air.

Air when taken into a compressor at atmospheric pressure and ordinary temperature, becomes very hot when it is compressed. As the example below will show, the temperature of atmospheric air of 60° F. rises to 484° F. when compressed to 100 pounds pressure, that is, it becomes about 150° hotter than steam at 100 pounds gauge pressure.

All air compressors are therefore equipped with a water-jacket for cooling the cylinder walls. By this cooling process and by radiation, the compressed air loses a considerable amount of heat and consequently volume.

The final temperature after compression is found by the following formula:

$$T_1 = T \times \left(\frac{P_1}{P}\right)^{0.29}$$

T_1 = absolute final temperature.
T = absolute initial temperature.
P_1 = absolute final pressure.
P = absolute initial pressure.

Example.—Air at atmospheric pressure and having a temperature of 60° F. is compressed to 100 pounds gauge pressure. What will be the final temperature of the compressed air?

Solution.—

$$T_1 = (60 + 460) \times \left(\frac{100 + 14.7}{14.7}\right)^{0.29} = 943.$$

The absolute final temperature is 943° F., which corresponds to a temperature of 943 — 460 or 483° F. This calculation requires the use of logarithms.

Loss Due to Altitude.

Though the pressure at sea level is 14.7 pounds per square inch, at higher altitudes, the atmospheric air is more rarified

and consequently exerts less pressure per square inch, as shown in the following table:

At ¼ mile above sea level.....14.2 lbs. per square inch.
At ½ mile above sea level.....13.33 lbs. per square inch.
At ¾ mile above sea level.....12.66 lbs. per square inch.
At 1 mile above sea level.....12.02 lbs. per square inch.
At 1¼ miles above sea level.....11.42 lbs. per square inch.
At 1½ miles above sea level.....10.88 lbs. per square inch.
At 2 miles above sea level..... 9.88 lbs. per square inch.

This table indicates an approximate reduction of ½ pound per square inch for every 1,000 feet of ascent. The losses at great altitudes, for which allowance should be made when buying an air compressor, are about 7% for ¼ mile, and 34% at two miles.

Clearance of Air Compressors.

Air and other gases cannot be pumped like liquids, as they expand, when the pressure is released, to the same extent as they had been compressed. If all or nearly all of the air is not discharged from the cylinder of the compressor, the remaining air expands to its initial pressure, and during this time prevents new air from entering the cylinder. It is evident that hereby the capacity of the compressor is greatly reduced, and it is therefore of great importance to have the least possible clearance in air compressors.

The clearance should not exceed 4%, and when buying an air compressor the percentage of clearance should be specified.

Lubrication.

Air compressors should be lubricated with graphite or a specially prepared oil. Glycerin should not be used, as it gums.

PUMPING WATER BY AIR COMPRESSOR.

If water is to be lifted from a well by means of air, it must be done in the proper manner, in order to obtain maximum quantity and maximum efficiency. For a *lift of 60 feet or less,* the air pipe should be submerged in the water to a distance of not less than two-thirds of the total length of the pipe. If, for instance, the water level in the well is 60 feet below the ground, then the air pipe should be $3 \times 60 = 180$ feet long. When the *height of water lift is greater than 60 feet,* the submersion may be some-

what less, but should never be less than 50% of the total length of pipe.

If the water sinks in the well, this can be observed by a decrease in the air pressure. The proper submersion of the pipe can always be calculated from this pressure. For instance, if the air pressure going to the well is 80 pounds gauge, then the submersion, assuming 10% friction loss, would be $80 \times 0.90 \times 2.31 = 166.3$ feet. The total length of the air pipe in this case should be not less than $2 \times 166.3 = 332.6$ feet.

If a well is equipped in this manner, maximum capacity is obtained with the least quantity of air.

Calculation of Quantity of Air Necessary.

In case of lifts of 60 feet or less, the amount of air necessary to lift a given quantity of water to a certain height can be determined by multiplying the quantity of water in gallons by the height in feet and dividing the product by 125. This will give the number of cubic feet of free air required per minute.

Example.—How many cubic feet of free air are necessary to lift 100 gallons of water per minute to a height of 60 feet?

Solution.—

$$\frac{100 \times 60}{125} = 48 \text{ cubic feet of free air are required.}$$

When the lift exceeds 60 feet, the amount of air required per gallon of water is very nearly 0.75 cubic feet. This does not vary much between lifts of 75 feet and 200 feet. The *air pressure* required for a given depth is equal to the head of submersion plus the friction head. The latter varies ordinarily between 8 and 12%, depending on the size and length of pipe, and may be considerably more if the pipe is extremely long and has many bends.

Calculation of Quantity of Water Lifted.

If it is desired to find how many gallons of water can be lifted by a given quantity of air, where the lift is 60 feet or less, the number of cubic feet of free air should be multiplied by the constant 125 and the product divided by the height to which the water is to be lifted.

Example.—How many gallons of water can be lifted to a height of 60 feet by 100 cubic feet of free air per minute?

Solution—

$\frac{100 \times 125}{60} = 208$ gallons of water can be lifted per minute.

Dimensions of Water and Air Pipes.

The velocity for water or other liquids is usually 300 feet per minute, though it may be as high as 500 feet, and for air or other gases 3,000 to 6,000 feet per minute. If possible, these velocities should not be exceeded, since the friction is proportionate to the square of the velocity, i. e., the friction is increased 4 times if the velocity is doubled, 9 times if tripled, etc., from which it can readily be seen that the velocity is best kept within practical limits.

The following rules for finding the relations between size of pipes, and velocity and quantity of liquid or gas moved through the pipes, are employed in average practice:

$$Q = \frac{V \times a}{144} \qquad G = \frac{V \times a}{19.25}$$

$$Q = \frac{V \times d^2 \times 0.7854}{144} \qquad G = \frac{V \times d^2 \times 0.7854}{19.25}$$

$$a = \frac{Q \times 144}{V} \qquad a = \frac{G \times 19.25}{V}$$

$$d = \sqrt{\frac{Q \times 144}{V \times 0.7854}} \qquad d = \sqrt{\frac{G \times 19.25}{V \times 0.7854}}$$

Q = cubic feet per minute to be moved through a pipe.
G = gallons per minute to be moved through a pipe.
V = velocity of liquid or gas in feet per minute.
a = area of pipe in square inches.
d = diameter of pipe in inches.

Example.—What size water and air pipes would be required if the figures of the preceding example are assumed, i. e., 208 gallons of water and 100 cubic feet of free air per minute?

Solution.—If the water is to be lifted 60 feet, then the air pipe must be 180 feet long, or $180 - 60 = 120$ feet submerged. Hence the air pressure required would be not less than the pressure corresponding to a 120 foot water column, which would be $\frac{14.7}{34} \times 120 = 52$ pounds. Allowing 10% friction, we obtain as the required air pressure $\frac{52}{0.90} = 58$ pounds, or approximately 60 pounds. The volume of air (100 cubic feet) is decreased in

the same ratio as the absolute pressure is increased (see Boyle-Mariotte Law). A gauge pressure of 60 pounds $= 60 + 15 = 75$ pounds absolute pressure; hence $\frac{75}{15} = 5$, which is the ratio of compression, and consequently the ratio of volume reduction; thus $100 \div 5 = 20$ cubic feet of compressed air are required to lift the 208 gallons of water per minute. Two hundred and eight gallons are 27.9 cubic feet, and the total quantity of air and water which must be delivered through the water pipe is therefore $27.9 + 20 = 47.9$ cubic feet. To find the diameter of the pipe necessary, the formula $d = \sqrt{\frac{Q \times 144}{V \times 0.7854}}$ may be employed, and by substituting, we obtain:

$$d = \sqrt{\frac{47.9 \times 144}{300 \times 0.7854}} = 5.4 \text{ inches, or a 5-inch diameter pipe.}$$

The air pipe, if a velocity of 3,000 feet per minute is assumed, would be

$$d = \sqrt{\frac{20 \times 144}{3{,}000 \times 0.7854}} = 1.1, \text{ or a 1-inch pipe.}$$

Flow of Water.

A somewhat detailed treatment on "Flow of Water" has been given in the chapter "Physics." A few remarks, however, may not be considered out of place in this chapter.

The flow of water from the same characteristic orifices varies directly as the area and as the square root of the height from which the water has fallen. The friction loss with a small opening is proportionally greater than with a large opening.

The theoretical velocity of falling water is expressed by the following formula:

$$V = \sqrt{2gh}, \text{ or } V = 8 \times \sqrt{h}.$$

V = velocity in feet per second.
g = gravity = 32.2.
h = height or head of water in feet.

On account of friction, which on the average is about 20%, the velocity in practice is less.

Example.—What is the velocity of water flowing through a short piece of pipe under a head of 150 feet?

Solution.—$V = 0.80 \times \sqrt{150 \times 32.2 \times 2} = 79.6$ feet per second.

For pipes of small diameter and considerable length, the following formula gives better results:

$$V = 2.313 \times \sqrt{\frac{h \times d}{f \times l + 0.125 \times d}}$$

l = length of pipe in feet (straight).
d = diameter of pipe in inches.
f = coefficient of friction (usually 0.025).

Example.—What is the velocity of water flowing through a 3-inch pipe 125 feet long, under a head of 150 feet?

Solution.—$V = 2.313 \times \sqrt{\frac{150 \times 3}{0.025 \times 125 + 0.125 \times 3}} = 26.43$ feet per second.

If it is desired to calculate the time it takes to empty the liquid of a vessel into another vessel through a pipe of certain diameter and length, and a certain head, the above formula can be used.

Example.—In what time will 10,000 gallons of water flow through a 3-inch pipe 125 feet long under a head of 8 feet.

Solution.—

$$V = 2.313 \times \sqrt{\frac{8 \times 3}{0.025 \times 125 + 0.125 \times 3}} = 6 \text{ feet per second,}$$

or 360 feet per minute.

The velocity of the liquid being 360 feet per minute, the quantity of liquid in gallons which will flow through a 3-inch pipe in one minute at a velocity of 360 feet per minute is:

$$G = \frac{V \times d^2 \times 0.7854}{19.25} = \frac{360 \times 3 \times 3 \times 0.7854}{19.25} = 132 \text{ gallons}$$

per minute; hence for 10,000 gallons the time required is $\frac{10{,}000}{132}$ = 76 minutes nearly.

If this liquid should flow through a larger pipe in 40 minutes, then it would require

$$d = \sqrt{\frac{G \times 19.25}{V \times 0.7854}} = \sqrt{\frac{10{,}000 \times 19.25}{40 \times 360 \times 0.7854}}$$

= 4.123 inches, or in practice a 4-inch pipe.

Refrigeration

Production of Refrigeration.

Refrigeration is the process of abstracting heat from, or cooling, a body. Where water or air, of a lower temperature than that desired in the body to be cooled, is at hand, such water or air may be used to do the cooling. If temperatures of 37° F. and above are desired, ice may be employed, but for temperatures near or below the freezing point of water, either freezing mixtures or refrigerating machines must be used.

Physical Laws Bearing on Refrigeration.

Before describing in detail the various methods of cooling, a brief explanation of the most important physical laws bearing on refrigeration will be given, though some of these have been explained in the chapter "Physics."

Latent Heat.—The heat which is necessary to change the state of aggregation of a substance, for instance, the heat used in melting a solid body, or in evaporating a liquid, or the heat employed in sublimation, cannot be measured by a thermometer. No rise in temperature occurs until all of the substance is either melted or evaporated, because the heat is used exclusively for performing internal work, i. e. for separating the molecules of the substance in question. Such heat, which becomes hidden, as it were, in the body, is called *latent heat*, and we speak of the *latent heat of vaporization* and the *latent heat of fusion*.

Heat can also become latent when doing external work, viz., when expanding a gas whose temperature is maintained constant during expansion. The heat that apparently disappears in performing the work of expanding the gas is called the *latent heat of expansion*.

Mariotte's or Boyle's Law.—This law states that the volume of a permanent gas, whose temperature remains con-

stant, varies inversely as its pressure. For instance, if the pressure is doubled, the volume of the gas is reduced to one-half, and if the pressure is increased 6 times, the volume will only be 1/6 of the original one, provided of course that the temperature remains constant.

Gay-Lussac's or Charles' Law.—When a permanent gas is heated while the pressure remains constant, the volume of the gas will increase in direct proportion with the absolute temperature (see page 56). For each degree Fahrenheit rise in temperature, the increase in volume is 1/492 of the volume at 32° F.

In the same manner, the volume of a gas is reduced 1/492 for each degree Fahrenheit the temperature is lowered.

The above laws hold good also for vapors, for instance, ammonia, carbon dioxide, steam, etc., when they are sufficiently superheated.

Absolute Zero and Absolute Temperature.—It has been stated above that the volume of a gas is reduced 1/492 with each degree Fahrenheit reduction in temperature. As the expansive force of the gas depends on the temperature, and decreases directly with it, it is evident that at a temperature of 492° F. below the freezing point of water, or 460° F. below zero, the gas cannot have any expansive force, that consequently the motion of the molecules has ceased and that there is no heat. This temperature is called the *absolute zero,* and serves as the lowest point on the so-called absolute scale. The absolute temperature of a body is found by adding 460 to the temperature determined according to the Fahrenheit thermometer scale.

The point of absolute zero has never been reached, though Dewar has succeeded during his experiments on the liquefaction of hydrogen in producing a temperature of 423° F. below zero, while Olszewski in trying to liquefy helium, the lightest gas known, reached as low a temperature as 443° F. below zero. Recently helium has been liquefied at a temperature of 454° F. below zero.

Critical Temperature, Pressure and Volume.—The *critical temperature* of a gas is that temperature above which no increase of pressure will cause liquefaction. All the so-called permanent gases are above their critical temperature, and they

can, therefore, not be liquefied by the application of pressure, unless their temperature is first reduced below the critical temperature. At this temperature the gas reaches a state at which the latent heat of evaporation is zero.

Critical Temperatures, Pressures and Boiling Points at Atmospheric Pressure of Various Substances.

Substance.	Critical Temperature ° F.	Critical Pressure Pounds Gauge.	Boiling Point ° F.
Acetic acid, CH_3COOH.....	+449	1,102	+244
Air	—220	559	—312
Ammonia, NH_3.....	+266	1,675	— 27
Amyl butyrate, $C_3H_7CO_2C_5H_{11}$.	+639	355	+349
Argon, A	—186	744	—305
Benzene, C_6H_6	+588	867	+176
Carbon bisulphide, CS_2	+527	1,129	+115
Carbon dioxide, CO_2.....	+ 88	1,088	—112
Carbon monoxide, CO.....	—219	522	—310
Cumol (Kopropyl benzene), $C_6H_5CH(CH_3)_2$	+637	452	+306
Ethane, C_2H_6	+ 96	1,102	—128
Ethyl acetate, $C_2H_3O_2C_2H_5$....	+464	606	+166
Ethyl alcohol, C_2H_5OH.....	+437	911	+172
Ethyl chloride, C_2H_5Cl.....	+365	779	+54.5
Ethyl ether, $(C_2H_5)_2O$.....	+392	536	+ 15
Ethylene, C_2H_4	+ 50	759	—152
Hydrogen, H	—389	294	— 42
Hydrogen selenide, H_2Se.....	+365	1,337	— 41.8
Methane, CH_4	—115.2	807	—263
Methyl chloride, CH_3Cl.....	.....	....	— 10
Methyl ether, $(CH_3)_2O$.....	.....	....	— 10
Nitrogen, N	—231	500	—315
Nitrous oxide, N_2O.....	+ 95	1,088	—128
Nitric oxide, NO.....	—135	1,029	—245
Oxygen, O	—182	765	—296
Propane, C_3H_8	+206.6	779	— 49
Propyl alcohol, C_3H_7OH.....	+493	767	+207
Sulphur dioxide, SO_2.....	+247	1,142	+ 14
Toluol (methyl benzene), $C_6H_5CH_3$	+577	573	+232
Water, H_2O	+689	2,925	+212

The pressure necessary for liquefaction at the critical temperature is called the *critical pressure,* and the volume of one pound of the gas at this point is called the *critical volume.* For its determination it is compared with the volume of one pound of the same gas at 32° F. and at atmospheric pressure.

The table on page 473 gives the critical temperatures and pressures, as well as the boiling points at atmospheric pressure, of various substances.

Unit of Refrigeration.

The unit of refrigeration in this country is the number of B. T. U. which must be abstracted from one ton of water of 32° F., in order to produce one ton (2,000 pounds) of ice of the same temperature. Since the latent heat of ice is 144, i. e., since 144 B. T. U. must be withdrawn from one pound of ice of 32° F. to produce one pound of water of 32° F., the unit of refrigeration is equal to $2{,}000 \times 144$, or, 288,000 B. T. U., this being called one ton of refrigeration. Formerly, 142 B. T. U. was considered as the latent heat of ice, and consequently one ton of refrigeration was taken as $2{,}000 \times 142 = 284{,}000$ B. T. U. Recently, however, the latent heat has been shown to be 144 B. T. U. The employment of 288,000 B. T. U. as the unit of refrigeration was suggested by the American Society of Mechanical Engineers in 1906, and this value has since then been generally accepted.

The specified time in which this amount of refrigeration must be done by a machine of one ton capacity is 24 hours. Thus, if a machine is said to have a refrigerating capacity of 50 tons, it means, that the machine can produce 50 tons of refrigeration in 24 hours, or can abstract $288{,}000 \times 50$ B. T. U. from the refrigerant in 24 hours. One ton of refrigeration (in 24 hours) is equivalent to the abstraction of $\frac{288{,}000}{24} = 12{,}000$ B. T. U. per hour, or $\frac{12{,}000}{60} = 200$ B. T. U. per minute.

Cooling with Ice.

In localities where ice is plentiful, and consequently cheap, as for instance in the northern parts of this country; refrigeration may be obtained by means of ice, provided no lower temperature than approximately 37° F. is required. This method

of cooling, which is applied in a number of industries in the northern states, is economical only for comparatively small plants.

When applied to *breweries,* the cooling may be effected by simply putting the ice between the vats. A more convenient method is to store the ice in large quantities in separate rooms, generally above the rooms to be cooled, both being connected by air ducts. These are arranged in such a manner that the warm air rising in the ducts will pass over the ice, and after having thus been cooled, will by its own weight sink again into the cellars. Circulation is thus maintained. The same method is also adopted in *ice storage houses.* The temperatures which can be produced and maintained in this manner vary from 37° to 40° F.

Large ice rooms need only be filled once a year, smaller ones must be filled twice a year.

Cooling with Brine.

Lower temperatures can be obtained if the refrigeration of the cooling rooms is effected by means of brine instead of ice.

In the absence of a refrigerating machine, this method consists of cooling salt brine by the addition of ice, and by circulating the cold brine by means of a pump through refrigerating coils in the cooling rooms.

The ice which is put into the brine melts, and in so doing absorbs heat, which it abstracts from the brine, lowering its temperature accordingly. Since each pound of ice adds a pound of water to the brine, it is evident that the salt solution gradually becomes weaker. To prevent its becoming too weak, it is necessary to strengthen the brine from time to time, which may be done by allowing part of the brine to flow away, and adding to the remainder the proper quantity of salt. On account of this loss, it is evident that during the course of a year, a considerable amount of salt is wasted.

This method is therefore not economical, nor is it convenient, and it is gradually being replaced by refrigerating machines. The latter are by far more economical, on account of the small cost of power and maintainance as compared to the large amount of daily labor required for handling the ice.

Freezing Mixtures.

When ice or snow is melted, a certain amount of heat is required, this heat being abstracted from the surroundings. The same happens with any substance when it is melted or dissolved without its chemical composition being changed. Thus, for instance, a mixture of snow or pounded ice and ordinary salt liquefies, and in consequence of the consumption of heat necessary to bring about liquefaction, the temperature is lowered.

When diluted sulphuric acid is poured on snow or ice, the latter at once begins to melt, and in so doing the temperature is reduced considerably, the heat withdrawn being utilized for liquefying the snow or ice.

Some substances, when dissolved in water, lower the temperature considerably. For instance, ammonium nitrate when dissolved in water abstracts from the latter so much heat that the temperature is reduced almost 50° F.

On the above principle is based the preparation and use of so-called freezing mixtures. Their employment, however, can only be a restricted one, since the expense connected with their use is high. It will, however, be of interest to note from the following table of freezing mixtures, the temperatures obtained by certain mixtures of various substances.

It is self-understood that these low temperatures can only be attained when the mixture does no cooling work; if the latter is the case, ice and chemicals must constantly be added to remove the heat.

Temperatures Attained by Freezing Mixtures.

Mixture.	Proportion by Weight.	Temperature Drops in Degrees F.	
		From	To
Sodium chloride Snow or pounded ice	1 1	+32°	0°
Sodium chloride Snow or pounded ice	2 1	+32°	— 5°
Sodium chloride Snow or pounded ice	1 3	+32°	0°
Ammonium nitrate Water	1 1	+50°	+ 3°

Temperatures Attained by Freezing Mixtures.

(Continued.)

Mixture.	Proportion by Weight.	Temperature Drops in Degrees F.	
		From	To
Sodium phosphate Nitric acid, dilute	9 4	+59°	+16°
Potassium nitrate Sal-ammoniac Water	1 1 1	+46°	—12°
Nitric acid, dilute Snow or pounded ice	1 1	+ 7°	—31°
Potash Snow or pounded ice	4 3	+32°	—35°
Sodium phosphate Sal-ammoniac Nitric acid, dilute	9 6 4	+55°	+21°
Sodium sulphate Ammonium nitrate Nitric acid, dilute	6 5 4	+50°	—13°
Sal-ammoniac Potassium nitrate Water	5 5 16	+50°	+ 7°
Sodium carbonate Ammonium nitrate Water	1 1 1	+50°	+ 7°
Sodium sulphate Nitric acid, dilute	3 2	+50°	— 2°
Sodium sulphate Potassium nitrate Sal-ammoniac Water	8 5 5 16	+50°	+ 5°
Sodium sulphate Muriatic acid	8 5	+50°	0°
Sodium sulphate Sulphuric acid, dilute	5 4	+50°	+ 3°
Sulphuric acid, dilute Nitric acid, dilute Snow or pounded ice	1 1 2	— 2°	—40°
Sulphuric acid, dilute Snow or pounded ice	1 1	+23°	—42°
Calcium chloride Snow or pounded ice	3 2	+32°	—27°
Calcium chloride Snow or pounded ice	2 1	+32°	—44°

REFRIGERATING MACHINES.

According to the nature of the medium employed, refrigerating machines are divided into two classes, viz:

1. *Machines employing air.*
   a. Vacuum machines.
   b. Air compression machines.
   c. Dense air compression machines.
2. *Machines using liquefying gases,* as ammonia, carbon dioxide, sulphur dioxide, etc. Ammonia machines are divided into two classes, viz:
   a. Compression machines.
   b. Absorption machines.

Vacuum Machines.

The principle of the vacuum machine is based upon the fact that when water is exposed to a vacuum it evaporates very rapidly, and at the same time, its temperature is lowered considerably. If the pressure on the surface of the water is below 0.181 inch of mercury, which is almost a perfect vacuum, the boiling point of the water will be below 32° F., the freezing point of water. In other words, water is frozen to ice by the rapid evaporation taking place in vacuo. It is necessary, however, to remove the water vapor produced during the process, as it would otherwise be impossible to maintain the required vacuum for any length of time. The water vapors are absorbed by strong sulphuric acid, which from time to time must be strengthened by boiling. Lead vessels are used for this, as all other available metals are attacked by the strong acid.

The machine, which must be very finely built in order to be able to maintain a high vacuum continually, consists of a freezing chamber connected to a vacuum pump. The mixture of gases (air and water-vapor) is passed through a vessel containing trays filled with strong sulphuric acid, before entering the vacuum pump. The latter really has to handle only air, which is discharged as useless.

The ice which is made directly by these machines has a milky appearance, but is as hard as glass, and from a hygienic standpoint is to be considered pure, for it is claimed that all the germs are killed by the strong vacuum. In order to obtain trans-

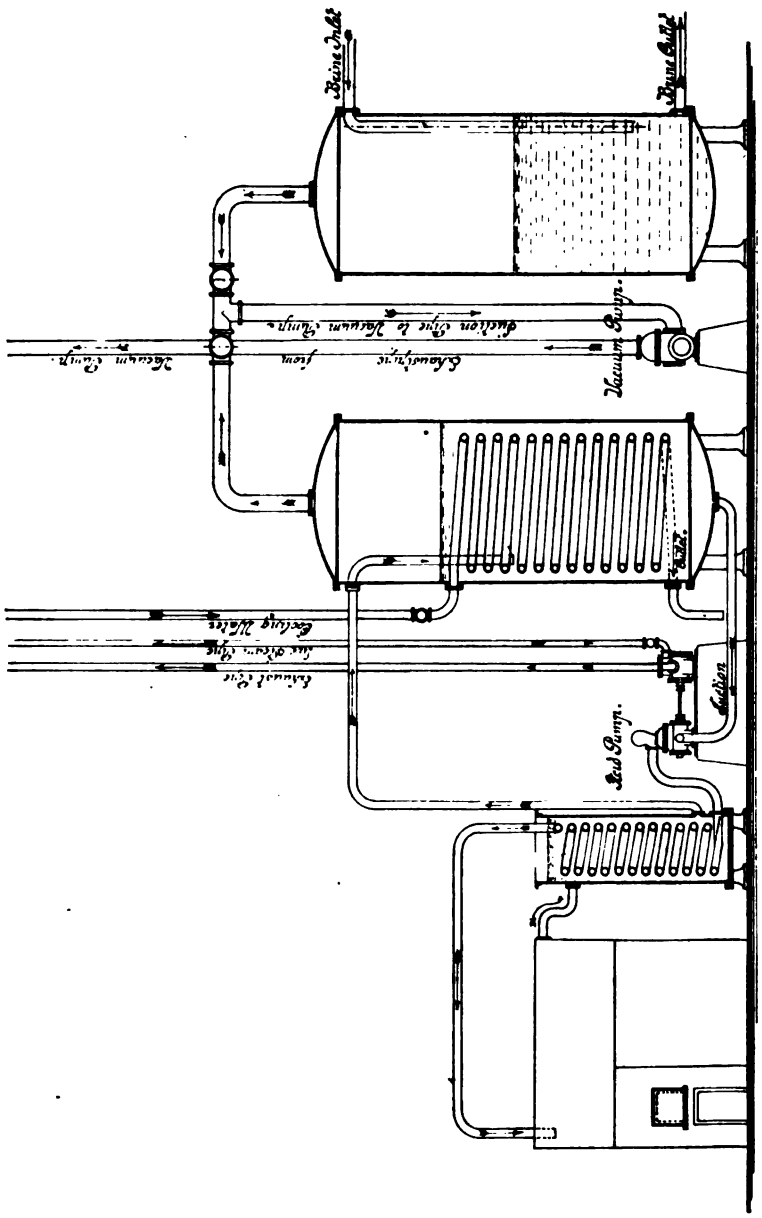


Figure 179. Vacuum Machine.

parent ice, it becomes necessary to make it indirectly. In this case, brine is reduced in temperature by evaporation through vacuum, and the ice is then frozen by the cold brine.

Cooling of Rooms by Vacuum Machine.—If rooms are to be cooled by means of the vacuum refrigerating machine, it must be done *indirectly,* i. e. brine must be cooled and circulated through the system. The cooling of rooms differs from ice-making only in that brine instead of water is put under a high vacuum.

A small quantity of the brine, usually about 1/6, is thereby evaporated, and part of this is absorbed by the sulphuric acid over which the vapors pass. The remaining part, including air, is drawn directly into the low-pressure vacuum pump, which discharges the mixture into the suction side of a smaller vacuum pump, the so-called high-pressure vacuum pump. An almost perfect vacuum is created in the low-pressure pump, since it works only against a very low pressure. It is claimed that a vacuum of $\frac{1}{1,500}$ atmosphere is continually created in the latest improved *Windhausen vacuum ice machine.* Evaporation of the brine is hastened by a stirring device with which the brine tank is equipped.

It is evident that constant evaporation must make the brine stronger, and in order to avoid the difficulties which would arise if the brine would become too strong, the system is equipped with a mixing tank, which the brine enters when coming from the cooling rooms. In this tank the quantity of water required, to make good the loss through evaporation, is added automatically. This is done by means of a float, which keeps the brine always at the same height, and consequently at the same strength.

The sulphuric acid which becomes gradually weaker, on account of its absorbing the vapors coming from the brine, is continually strengthened. The weak acid passes into a vessel where it is concentrated by boiling, and the strong acid flows out at the bottom, through the *heat exchanger,* where it is cooled by meeting the weak acid on its way to the *concentrator,* and back to the *absorber.*

The above short description of the construction and operation of the vacuum machine will suffice, since the efficiency of

even the latest machines of this type is far below that of the ammonia, carbon dioxide or sulphur dioxide machine.

Air Compression Machines.

The construction of the air compression machine is based upon the fact that when air is compressed, heat is developed, and when air is allowed to expand, heat is absorbed. The latter is only true, however, in case the air performs work while expanding, for instance when the expansion takes place in a cylinder, the piston of which is pushed by the expanding gas. Air under pressure, leaving a large vessel through a small opening, will not do any cooling, except a trifle at the orifice, and this only because it performs some work there, which, however, is extremely small in comparison with the work it could perform if expanding behind a piston. The work performed in this instance is the friction of the out-rushing air on the sides of the opening, and the work required to push the outer air aside. At still higher pressures, Prof. Linde found that an appreciable cooling effect is exercised by a reduction of pressure, the temperature being reduced about ½° F. per atmosphere (15 pounds) pressure.

After the air which has been taken into the compressor from the atmosphere has been compressed to the desired point, it is delivered into a cooler over which water is showered, and cooled as low as is possible with the water. It is then allowed to flow into a second cylinder, the so-called expansion cylinder, in which the air expands, at the same time driving the piston through the cylinder. At the end of the stroke, the air is considerably cooler, and is made to enter the rooms to be cooled. The compression and the expansion cylinders are coupled to the same shaft, and the work performed in the expansion cylinder by the expanding air materially assists the compressor.

It is evident that the heat produced by compressing, and then removed by the water in the cooler, must be equal in amount to the heat transformed into work during the expansion of the air, or, in other words, if we ascertain how many heat-units the water has received, by multiplying the number of pounds of water by the difference in temperature in degrees F. of the water when entering and when leaving the cooler, we have the amount of cooling work which the expanded air can do.

This cooling work amounts to about 1,450 B. T. U. per engine horse-power, which corresponds to a heat abstraction from the refrigerating room of 1 B. T. U. for every 20 B. T. U. furnished to the steam engine, if the latter consumes 30 pounds of steam per H. P. per hour. Under the same conditions, a well-built ammonia machine abstracts 1 B. T. U. for every 3 B. T. U. furnished to the engine, which shows that the latter refrigerating machine is almost 7 times as economical as an air machine. The volume of air which must be compressed is 27 times larger than the volume of ammonia necessary for the production of the same amount of refrigeration. Air compression machines are now but little used.

Dense-Air Compression Machines.

The principle of the dense-air compression machine is, in the main, the same as that of the one just described. The only difference is the fact that the dense-air machine admits the air to the compressor at a higher pressure than atmospheric, generally at about 60 pounds gauge. This is done in order that the size of the compressor may be reduced.

The construction of the machine is the same as that of the machine taking in air at atmospheric pressure, except that the expanded air is sent into coils of pipes instead of into the room directly. This is done in order to be able to maintain a pressure of 60 pounds.

The following is a graphical sketch of the *Allen Dense-Air System* (see Figure 180).

The letter A designates the steam-cylinder, which drives the apparatus; B is the air compressor, which is surrounded by a water-jacket; C is a copper coil submerged in a water-bath.

The air compressed in B is cooled by water almost to the temperature of the water flowing over the copper coil. It is then further cooled in the return air cooler, D, through which flows the cold air returning from the refrigerating room. From here the cooled and compressed air enters the expansion-cylinder E. When it has filled this cylinder to about one-third of its volume, the air is shut off and then drives the piston to the end of the cylinder. Thereby, the pressure of the air decreases to ⅓ of its original pressure, and its temperature is thus lowered the same

amount it was heated during compression, plus the lowering of temperature caused by the cooling with water and return air. Before the expanded and cold air enters the conveying pipe it passes through the trap F, which collects the lubricating oil used in the cylinder, and also the small amount of snow due to the freezing of the moisture in the air. The air thus enters the cooling system in a pure and clear condition. From time to time it becomes necessary to rid the trap from the accumu-

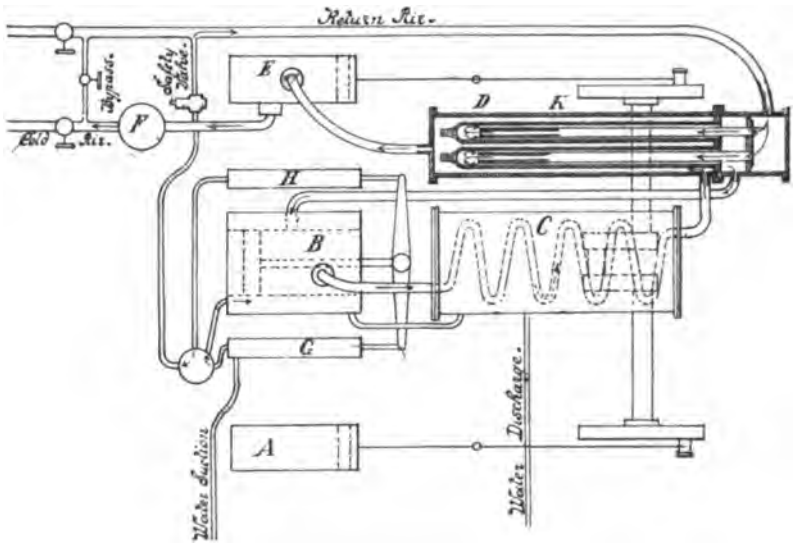


Figure 180. Allen Dense Air System.

lated snow, and for this purpose it is provided with a heating pipe. Once in 12 or 24 hours the trap must be warmed by means of this pipe, and the deposits withdrawn. The machine is arranged at the same time to thaw out and blow into the trap any possible frozen deposits from the expansion-cylinder.

G is a common plunger water pump, which supplies the cooling water for the trap K, for the water-bath surrounding the copper coil C, and for the water-jacket of the compression cylinder B.

H is a small supplementary air pump for making up the loss of air through stuffing boxes, thereby enabling the machine to maintain the same pressure continually.

K is a small trap which extracts moisture from the newly supplied air.

The dense-air machine is more economical and gives less trouble than the ordinary air compression machine, since the air is used over and over again, and can consequently be almost entirely deprived of its moisture, which latter is the most troublesome factor in open cycle air machines. All air machines, however, no matter how ingeniously constructed, can not compete with the modern ammonia, carbon dioxide or sulphur dioxide machines. There is, therefore, no reason for giving a more detailed description of their construction and operation.

Compression Machines Using Liquefying Gases.

In general, it may be said that the principle of the compression machine is based on the ability of certain gases to liquefy on being cooled with water or air, when compressed to pressures practically obtainable, and on the other hand their ability to evaporate at the temperatures required in the cooling coil. The heat necessary for evaporation is abstracted from the surroundings, i. e., the air of the cooling rooms, brine, etc., and the temperature of these is consequently lowered.

Process of Producing Refrigeration.—The process of producing continual refrigeration is practically the same for all compression machines. There is required, first, the refrigerating medium, i. e., the liquefied vapor, secondly, the ice machine, which compresses and circulates the refrigerating medium, and thirdly, the cooling fluid, generally water, which assists in liquefying the vapor by carrying away not only the heat which the refrigerating medium absorbed in the refrigerator, but also the heat generated by the compression of the gas in the compressor.

The efficiency of the ice machine is greatest when the difference between the temperatures at which the refrigerating medium takes up heat in the refrigerator, and at which it gives it off again in the condenser, is the smallest.

Figure 181 gives a diagrammatical sketch of a compression machine.

The compressor A serves the purpose of compressing the vapor, which enters through the suction valve H at the evaporating pressure and temperature, and is forced out through the

discharge valve and discharge pipe C, at the condenser pressure and temperature into the condenser D. This consists of a series of pipes arranged horizontally one above the other, water being continually showered over them. The temperature of the cooling water must be lower than that of the compressed vapor, in order that the latent heat of the latter may be absorbed. The vapor being consequently deprived of its heat will at last liquefy. In this condition it is admitted to the refrigerator F, through the ex-

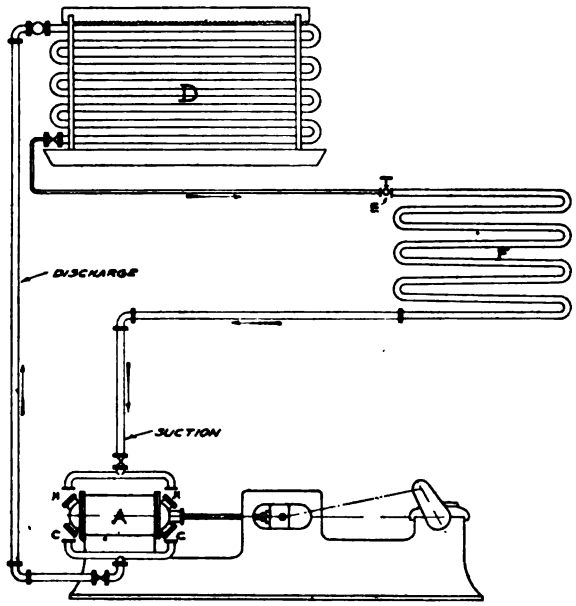


Figure 181. Compression System.

pansion valve E, which is so constructed that it allows only small quantities of the liquid to pass through, and that the flow can be very finely regulated.

The refrigerator F consists of pipe coils, which are either placed in the room to be cooled, or are submerged in brine, which, after having been cooled, serves as refrigerating medium.

The liquefied vapor, which circulates through the refrigerator coils, and whose temperature is lower than that of its surroundings, absorbs heat from the latter until it finally evaporates almost completely, and the vapor reaches the suction valve of the compressor in a saturated condition.

Compression Machine and Heat Engine Compared.—A compression machine may be regarded as a reversed heat engine. In the latter, heat is transformed into work, and if this heat engine drives a compressor, the work is transformed back into heat. In the steam engine, the steam enters the cylinder at a high pressure and temperature, expands, and leaves the cylinder at a low pressure and temperature; in the compression machine, the vapor enters the compressor at a low temperature and pressure, is compressed, and leaves the cylinder at a high temperature and pressure.

General Requirements of Refrigerating Media.—In order to give the best results, the refrigerating medium must possess certain qualifications, viz., it should be able to absorb and give off the greatest possible quantity of heat in one cycle of operation of the machine with the least expenditure of energy; it should possess a low boiling point, high latent heat, high critical temperature, high condensing temperature and low condensing pressure; and it should furthermore be non-corrosive on metals, non-explosive, low in price, a self-lubricant, and non-decomposing at moderate temperature.

Ammonia, carbon dioxide, and sulphur dioxide, are so far the substances which come nearest to these requirements. Other substances which have been used include sulphuric or ethyl ether, methyl ether, ethyl chloride, and methyl chloride.

Methods of Compression.

Compression may be performed in any one of the following three different ways:

1. Dry compression.
2. Wet compression.
3. Oil circulation.

Dry Compression.—The gas is compressed in a jacketed compressor, and water is circulated through this jacket, to remove, as much as possible, the heat produced by the compression.

Wet Compression.—With the gas, a small portion of liquid enters the compressor, which is all or partly evaporated during the compression period, whereby the same effect is accomplished as with the water jacket in the dry compressor, but to a somewhat higher degree. A comparison between wet and dry compression is given on pages 552 and 553.

Oil Circulation.—This method is gradually going out of use. It is briefly described on page 509.

THE AMMONIA COMPRESSION MACHINE.

Of all the liquefiable gases which can be used in compression machines, ammonia is the one which most nearly meets the various requirements. For this reason, it is most commonly employed for refrigerating purposes. The properties of ammonia and the description of the ammonia compression machine, will therefore be given first.

Properties of Ammonia.

On pages 490 to 493 is given a table of properties of saturated ammonia vapor. This table is one worked out recently (1912) by H. J. Macintire, copyrighted by the Ice Trade Journal Co., New York, and it has been adopted as the standard in this book. Another recent table* worked out by G. A. Goodenough and W. E. Mosher gives data not much at variance with the Macintire table. Both give values considerably different from those of De Volson Wood's table. For method of interpolating the table, see page 808.

Various properties of liquid, superheated and saturated ammonia are given below. Some of the other chemical properties of ammonia, that have no direct bearing on refrigeration, are given in the chapter "Chemistry."

Specific Heat of Liquid Ammonia.—On the average, the specific heat of liquid ammonia may for all practical calculations be safely taken as 1.1. It varies with the temperature, and different authors give values which are widely divergent. For calculating the specific heat at any temperature, various formulas have been given. Ledoux' equation, with a new constant derived by Elleau and Ennis from experiments, and adapted to the Fahrenheit scale, follows:

$$c = 0.9184 + 0.002032 \times t.$$

c = specific heat of liquid ammonia.

t = temperature of the liquid in degrees Fahrenheit.

The latest formula for calculating specific heat is that of Dieterici, the temperature here being also expressed in Fahrenheit, as follows:

$$c = 1.124 + 0.00119 \times (t - 32).$$

* Power, Vol. 38, No. 2, July 8, 1913.

Though the specific heat of liquid ammonia has not yet been definitely determined, this latter formula may be considered as the most accurate, and will be used as the basis in this book.

Example.—What is the specific heat of liquid ammonia at 86° F?

Solution.—According to Dieterici's formula,

$$c = 1.124 + 0.00119 \times (86 - 32) = 1.188.$$

The specific heat of liquid ammonia at 86°F. is 1.188.

Specific Heat of Saturated and Superheated Ammonia.—The specific heat of saturated ammonia vapor is negative for all temperatures less than 555°F. absolute, which means that the saturated vapor will condense with adiabatic expansion, giving up its heat to the remaining vapor, thereby maintaining the temperature corresponding to the pressure. When compressed, heat must be abstracted, otherwise it will become superheated. The specific heat of ammonia gas in the superheated state, is given by Regnault as 0.50836 at constant pressure, and 0.38425* at constant volume.

Specific Gravity of Liquid Ammonia.—The specific gravity of liquid ammonia at any temperature may be accurately expressed by D'Andreff's formula:

$$S = 0.6502 - 0.000777 \times t.$$

S = specific gravity.

t = temperature in degrees Fahrenheit.

Example.—What is the specific gravity, and density in pounds per cubic foot, of liquid ammonia at 75° F?

Solution.—$S = 0.6502 - 0.000777 \times 75 = 0.591925.$

The specific gravity is 0.591925. The density, or weight of a cubic foot of liquid ammonia, at 75° F., is $0.591925 \times 62.425 = 36.95$ pounds.

Volume of Liquid Ammonia.—The volume of liquid ammonia in cubic feet per pound, commonly called *specific volume,* is the reciprocal of the density in pounds per cubic foot as obtained above, and may be calculated by the following formula:

$$V = \frac{1}{(0.6502 - 0.000777 \times t) \times 62.425}$$

* Frequently given as 0.3913.

At a temperature of 75° F., for instance, the specific volume of liquid ammonia would be:

$$V = \frac{1}{(0.6502 - 0.000777 \times 75) \times 62.425} = 0.027.$$

Coefficient of Expansion of Liquid Ammonia.—The coefficient of expansion of liquid ammonia is 0.00204.

Volume and Weight of Saturated Ammonia Vapor.—These figures, which vary considerably with the pressure (and temperature), are found in all ammonia tables. At atmospheric pressure, the volume of one pound of saturated ammonia, generally called specific volume, is 17.99 cubic feet, and the weight of 1 cubic foot is 0.0555 pound. The two values are of course in all cases, reciprocals of each other, that is, the weight equals $\frac{1}{\text{volume}}$, and the volume equals $\frac{1}{\text{weight}}$.

Specific Gravity of Superheated Ammonia.—The specific gravity of ammonia gas as compared with air is 0.596.

Relation between Temperature and Pressure.—The relation between temperature and pressure of saturated ammonia vapor is given in the ammonia table. At atmospheric pressure, the temperature is — 27° F.

Latent Heat of Vaporization.—The latent heat of vaporization of ammonia decreases with increasing pressure, and its values at the different pressures are given in the ammonia table. At atmospheric pressure, it is 588.50 B. T. U., and at 0° F., it is 569.886 B. T. U.

Critical Temperature and Pressure.—The critical temperature of ammonia gas is 266° F., and the critical pressure at this point amounts to 1,675 pounds per square inch.

Solubility.—The solubility of ammonia in water varies with the temperature of the water and with the pressure, and is the greater the lower the temperature and the higher the pressure. A 32° F. and atmospheric pressure one volume of water absorbs about 1,180 volumes of ammonia gas. Tables of solubilities are given on pages 568 and 569.

Behavior towards Metals.—Ammonia does not affect any metals except brass and copper, but these are rapidly destroyed.

Properties of Saturated Ammonia.*

Temperature.		Pressure in Pounds per Square Inch.		Saturated Vapor.		Liquid.	Total Heat above 32 Degrees Fahrenheit B. T. U.		
Degrees Fahr.	Absolute.	Absolute.	Gauge.	Volume in Cubic Feet per Lb.	Weight in Lbs. per Cubic Foot.	Weight in Lbs. per Cubic Foot.	Of the Liquid.	Of Dry and Saturated Vapor.	Total Latent Heat in B. T. U. per Lb.
—40	420	9.25	—5.45	28.06	.0356	45.74	—77.80	519.50	597.30
—39	421	9.68	—5.02	27.16	.0369	.65	—76.76	519.90	596.66
—38	422	9.96	—4.74	26.26	.0381	.57	—75.72	520.30	596.02
—37	423	10.24	—4.46	25.37	.0395	.48	—74.68	520.70	595.38
—36	424	10.67	—4.03	24.48	.0409	.39	—73.63	521.09	594.72
—35	425	11.10	—3.60	23.59	.0425	.21	—72.58	521.48	594.06
—34	426	11.52	—2.18	22.83	.0438	.12	—71.54	521.87	593.41
—33	427	11.95	—2.75	22.08	.0452	45.04	—70.49	522.25	592.74
—32	428	12.37	—2.33	21.33	.0469	44.95	—69.44	522.63	592.07
—31	429	12.80	—1.90	20.58	.0486	.87	—68.39	523.01	591.40
—30	430	13.32	—1.38	19.83	.0505	.78	—67.34	523.39	590.73
—29	431	13.73	—0.97	19.30	.0518	.69	—66.29	523.77	590.06
—28	432	14.08	—0.62	18.77	.0531	.60	—65.24	524.16	589.40
—27	433	14.50	—0.20	18.24	.0549	.52	—64.18	524.55	588.73
—26	434	15.08	+0.38	17.71	.0565	.44	—63.13	524.94	588.07
—25	435	15.50	+0.70	17.19	.0582	44.35	—62.07	525.32	587.39
—24	436	15.94	+1.24	16.75	.0598	44.27	—61.01	525.71	586.72
—23	437	16.37	1.67	16.31	.0613	44.18	—59.96	526.10	586.06
—22	438	16.92	2.22	15.81	.06328	44.10	—58.896	526.482	585.378
—21	439	17.41	2.71	15.40	.06498	44.01	—57.836	526.870	584.706
—20	440	17.91	3.21	15.00	.06671	43.92	—56.775	527.256	584.031
—19	441	18.41	3.71	14.61	.06847	.84	—55.713	527.640	583.353
—18	442	18.92	4.22	14.24	.07026	.75	—54.650	528.022	582.672
—17	443	19.44	4.74	13.88	.07208	.67	—53.586	528.402	581.988
—16	444	19.97	5.27	13.53	.07393	.58	—52.521	528.780	581.301
—15	445	20.51	5.81	13.19	.07582	43.49	—51.455	529.156	580.611
—14	446	21.06	6.36	12.86	.07774	.41	—50.388	529.530	579.918
—13	447	21.62	6.92	12.55	.07968	.33	—49.320	529.902	579.222
—12	448	22.19	7.49	12.24	.08166	.24	—48.250	530.272	578.522
—11	449	22.77	8.07	11.95	.08368	.16	—47.179	530.640	577.819
—10	450	23.36	8.66	11.66	.08574	43.07	—46.107	531.006	577.113
— 9	451	23.96	9.26	11.38	.08784	42.99	—45.034	531.370	576.404
— 8	452	24.58	9.88	11.11	.08998	.91	—43.960	531.732	575.692
— 7	453	25.21	10.51	10.85	.09216	.83	—42.885	532.092	574.977
— 6	454	25.85	11.15	10.60	.09437	.75	—41.809	532.450	574.259
— 5	455	26.50	11.80	10.35	.09662	42.67	—40.732	532.806	573.538
— 4	456	27.16	12.46	10.11	.09891	.58	—39.654	533.160	572.814
— 3	457	27.84	13.14	9.878	.1012	.50	—38.575	533.512	572.087
— 2	458	28.54	13.84	9.654	.1036	.42	—37.495	533.862	571.357
— 1	459	29.26	14.56	9.436	.1060	.34	—36.413	534.210	570.623

* Table by H. J. Macintire, copyright by Ice Trade Journal Co., New York.

Properties of Saturated Ammonia.

Temperature.		Pressure in Pounds per Square Inch.		Saturated Vapor.		Liquid.	Total Heat above 32 Degrees Fahrenheit B. T. U.		
Degrees Fahr.	Absolute.	Absolute.	Gauge.	Volume in Cubic Feet per Lb.	Weight in Lbs. per Cubic Foot.	Weight in Lbs. per Cubic Foot.	Of the Liquid.	Of Dry and Saturated Vapor.	Total Latent Heat in B. T. U. per Lb.
0	460	29.99	15.29	9.223	.1084	42.26	—35.330	534.556	**569.886**
1	461	30.74	16.04	9.015	.1109	.18	—34.246	534.900	**569.146**
2	462	31.50	16.80	8.812	.1134	.10	—33.161	535.242	**568.403**
3	463	32.27	17.57	8.614	.1160	42.02	—32.075	535.582	**567.657**
4	464	33.06	18.36	8.421	.1187	41.95	—30.987	535.920	**566.907**
5	465	33.86	19.16	8.233	.1214	41.87	—29.898	536.256	**566.154**
6	466	34.67	19.97	8.051	.1241	.79	—28.807	536.591	**565.398**
7	467	35.49	20.79	7.874	.1269	.71	—27.715	536.924	**564.639**
8	468	36.33	21.63	7.702	.1297	.64	—26.622	537.256	**563.878**
9	469	37.18	22.48	7.535	.1326	.56	—25.527	537.586	**563.113**
10	470	38.05	23.35	7.372	.1356	41.48	—24.431	537.914	**562.345**
11	471	38.93	24.23	7.213	.1386	.41	—23.334	538.240	**561.574**
12	472	39.83	25.13	7.058	.1416	.34	—22.235	538.564	**560.799**
13	473	40.75	26.05	6.907	.1447	.26	—21.135	538.887	**560.022**
14	474	41.68	26.98	6.759	.1479	.18	—20.034	539.208	**559.242**
15	475	42.63	27.93	6.616	.1511	41.11	—18.931	539.527	**558.458**
16	476	43.60	28.90	6.476	.1543	41.03	—17.827	539.844	**557.671**
17	477	44.59	29.89	6.340	.1576	40.96	—16.721	540.158	**556.879**
18	478	45.60	30.90	6.208	.1610	.88	—15.614	540.470	**556.084**
19	479	46.63	31.93	6.079	.1644	.81	—14.506	540.780	**555.286**
20	480	47.68	32.98	5.953	.1679	40.73	—13.397	541.088	**554.485**
21	481	48.75	34.05	5.830	.1715	.66	—12.287	541.394	**553.681**
22	482	49.83	35.13	5.709	.1751	.59	—11.175	541.697	**552.872**
23	483	50.93	36.23	5.590	.1788	.52	—10.062	541.998	**552.060**
24	484	52.05	37.35	5.475	.1825	.44	— 8.949	542.296	**551.245**
25	485	53.19	38.49	5.363	.1863	40.37	— 7.835	542.591	**550.426**
26	486	54.35	39.65	5.255	.1902	.30	— 6.719	542.884	**549.603**
27	487	55.53	40.83	5.150	.1941	.23	— 5.602	543.174	**548.776**
28	488	56.73	42.03	5.047	.1981	.16	— 4.484	543.462	**547.946**
29	489	57.95	43.25	4.946	.2022	.09	— 3.365	543.748	**547.113**
30	490	59.19	44.49	4.847	.2063	40.01	— 2.245	544.031	**546.276**
31	491	60.45	45.75	4.750	.2105	39.94	— 1.123	544.312	**545.435**
32	492	61.73	47.03	4.655	.2148	.87	0.0	544.590	**544.590**
33	493	63.03	48.33	4.563	.2192	.80	1.125	544.866	**543.741**
34	494	64.35	49.65	4.473	.2236	.73	2.251	545.139	**542.888**
35	495	65.70	51.00	4.385	.2281	39.66	3.378	545.410	**542.032**
36	496	67.07	52.37	4.300	.2326	.60	4.507	545.678	**541.171**
37	497	68.46	53.76	4.216	.2372	.53	5.637	545.944	**540.307**
38	498	69.87	55.17	4.134	.2419	.45	6.768	546.208	**539.440**
39	499	71.31	56.61	4.054	.2467	.38	7.901	546.469	**538.568**
40	500	72.77	58.07	3.976	.2515	39.31	9.035	546.728	**537.693**
41	501	74.25	59.55	3.900	.2564	.25	10.170	546.984	**536.814**

Properties of Saturated Ammonia.

Temperature.		Pressure in Pounds per Square Inch.		Saturated Vapor.		Liquid.	Total Heat above 32 Degrees Fahrenheit B. T. U.		
Degrees Fahr.	Absolute.	Absolute.	Gauge.	Volume in Cubic Feet per Lb.	Weight in Lbs. per Cubic Foot.	Weight in Lbs. per Cubic Foot.	Of the Liquid.	Of Dry and Saturated Vapor.	Total Latent Heat in B. T. U. per Lb.
42	502	75.76	61.06	3.826	.2614	39.18	11.307	547.238	**535.931**
43	503	77.30	62.60	3.753	.2665	.11	12.445	547.490	**535.145**
44	504	78.86	64.16	3.681	.2716	.04	13.584	547.739	**534.155**
45	505	80.45	65.75	3.612	.2768	38.98	14.725	547.985	**533.260**
46	506	82.06	67.36	3.544	.2821	.91	15.867	548.229	**532.362**
47	507	83.70	69.00	3.477	.2875	.85	17.010	548.471	**531.461**
48	508	85.36	70.66	3.412	.2930	.79	18.155	548.710	**530.555**
49	509	87.05	72.35	3.349	.2986	.71	19.301	548.946	**529.645**
50	510	88.76	74.06	3.287	.3042	38.64	20.448	549.180	**528.732**
51	511	90.49	75.79	3.226	.3100	.57	21.597	549.411	**527.814**
52	512	92.24	77.54	3.167	.3158	.51	22.747	549.639	**526.892**
53	513	94.01	79.31	3.109	.3217	.44	23.899	549.864	**525.965**
54	514	95.81	81.11	3.052	.3277	.38	25.052	550.087	**525.035**
55	515	97.64	82.94	2.996	.3338	38.31	26.206	550.307	**524.101**
56	516	99.50	84.80	2.941	.3400	.25	27.361	550.524	**523.163**
57	517	101.38	86.68	2.887	.3463	.18	28.517	550.739	**522.222**
58	518	103.29	88.59	2.835	.3527	.12	29.675	550.951	**521.276**
59	519	105.23	90.53	2.784	.3592	.05	30.834	551.160	**520.326**
60	520	107.20	92.50	2.734	.3658	37.99	31.993	551.367	**519.374**
61	521	109.21	94.51	2.685	.3724	.92	33.153	551.571	**518.418**
62	522	111.26	96.56	2.637	.3792	.86	34.314	551.773	**517.459**
63	523	113.34	98.64	2.590	.3861	.80	35.477	551.972	**516.495**
64	524	115.45	100.75	2.544	.3931	.73	36.641	552.168	**515.527**
65	525	117.59	102.89	2.499	.4002	37.67	37.806	552.361	**514.555**
66	526	119.76	105.06	2.455	.4074	.61	38.972	552.551	**513.579**
67	527	121.96	107.26	2.411	.4147	.55	40.140	552.739	**512.599**
68	528	124.20	109.50	2.369	.4221	.48	41.310	552.924	**511.614**
69	529	126.44	111.74	2.328	.4296	.42	42.482	553.105	**510.623**
70	530	128.72	114.02	2.288	.4372	37.36	43.656	553.283	**509.627**
71	531	131.03	116.33	2.248	.4449	.29	44.831	553.459	**508.628**
72	532	133.37	118.67	2.209	.4527	.23	46.008	553.632	**507.624**
73	533	135.74	121.04	2.171	.4606	.17	47.186	553.802	**506.616**
74	534	138.14	123.44	2.133	.4686	.11	48.366	553.969	**505.603**
75	535	140.57	125.87	2.097	.4768	37.03	49.547	554.133	**504.586**
76	536	143.03	128.33	2.061	.4851	36.99	50.729	554.295	**503.566**
77	537	145.52	130.82	2.026	.4935	.93	51.912	554.454	**502.542**
78	538	148.04	133.34	1.992	.5020	.86	53.097	554.611	**501.514**
79	539	150.60	135.90	1.959	.5106	.80	54.283	554.766	**500.483**
80	540	153.19	138.49	1.926	.5193	36.74	55.470	554.918	**499.448**
81	541	155.82	141.12	1.894	.5281	.68	56.659	555.067	**498.408**
82	542	158.49	143.79	1.862	.5370	.62	57.849	555.213	**497.364**

Temperature.		Pressure in Pounds per Square Inch.		Saturated Vapor.		Liquid.	Total Heat above 32 Degrees Fahrenheit B. T. U.		
Degrees Fahr.	Absolute.	Absolute.	Gauge.	Volume in Cubic Feet per Lb.	Weight in Lbs. per Cubic Foot.	Weight in Lbs. per Cubic Foot.	Of the Liquid.	Of Dry and Saturated Vapor.	Total Latent Heat in B. T. U. per Lb.
83	543	161.19	146.49	1.831	.5460	36.56	59.040	555.356	**496.316**
84	544	163.93	149.23	1.801	.5551	.50	60.233	555.496	**495.263**
85	545	166.70	152.00	1.772	.5643	36.44	61.427	555.633	**494.206**
86	546	169.50	154.80	1.743	.5737	.39	62.622	555.768	**493.146**
87	547	172.34	157.64	1.714	.5832	.33	63.819	555.900	**492.081**
88	548	175.22	160.52	1.686	.5928	.27	65.017	556.029	**491.012**
89	549	178.13	163.43	1.659	.6025	.21	66.216	556.155	**489.939**
90	550	181.08	166.38	1.633	.6123	36.15	67.417	556.278	**488.861**
91	551	184.07	169.37	1.607	.6222	.10	68.619	556.398	**487.779**
92	552	187.09	172.39	1.581	.6323	.04	69.822	556.515	**486.693**
93	553	190.15	175.45	1.556	.6425	35.98	71.027	556.629	**485.602**
94	554	193.20	178.50	1.532	.6529	.92	72.233	556.740	**484.507**
95	555	196.40	181.70	1.508	.6634	35.86	73.440	556.848	**483.408**
96	556	199.59	184.89	1.484	.6741	.81	74.649	556.954	**482.305**
97	557	202.82	188.12	1.460	.6849	.75	75.859	557.057	**481.198**
98	558	206.09	191.39	1.437	.6959	.70	77.070	557.157	**480.087**
99	559	209.40	194.70	1.415	.7070	.63	78.283	557.254	**478.971**
100	560	212.75	198.05	1.392	.7183	35.58	79.497	557.348	**477.851**
101	561	216.15	201.45	1.370	.7297	.52	80.712	557.439	**476.727**
102	562	219.59	204.89	1.349	.7413	.46	81.929	557.527	**475.598**
103	563	223.07	208.37	1.328	.7530	.41	83.147	557.612	**474.465**
104	564	226.60	211.90	1.307	.7650	.35	84.366	557.694	**473.328**
105	565	229.5	214.8	1.297	.770	35.28	85.58	557.89	**472.31**
106	566	232.5	217.8	1.279	.783	.21	86.80	557.98	**471.18**
107	567	235.8	221.1	1.261	.794	.15	88.03	558.07	**470.04**
108	568	239.1	224.4	1.242	.816	.10	89.26	558.16	**468.90**
109	569	242.4	227.7	1.224	.828	.02	90.48	558.25	**467.77**
110	570	246.0	231.3	1.205	.830	34.98	91.71	558.33	**466.62**
111	571	249.6	234.9	1.188	.843	.94	92.93	558.40	**465.49**
112	572	253.1	238.4	1.170	.856	.89	94.16	558.50	**464.34**
113	573	256.6	241.9	1.153	.869	.83	95.38	558.58	**463.20**
114	574	260.2	245.5	1.135	.882	.78	96.61	558.66	**462.15**
115	575	264.4	249.7	1.118	.895	34.74	97.84	558.74	**460.90**
116	576	269.1	254.4	1.101	.909	.68	99.07	558.77	**459.70**
117	577	273.0	258.3	1.084	.924	.62	100.31	558.80	**458.49**
118	578	276.9	262.2	1.067	.938	.56	101.55	558.83	**457.28**
119	579	281.2	266.5	1.050	.953	.50	102.79	558.86	**456.07**
120	580	285.1	270.4	1.033	.968	34.43	104.03	558.89	**454.86**
121	581	289.6	274.9	1.018	.983	.39	105.27	558.91	**453.64**
122	582	294.0	279.3	1.002	0.999	.34	106.52	558.93	**452.41**
123	583	298.6	283.9	0.987	1.014	.29	107.76	558.95	**451.19**
124	584	302.9	287.2	0.972	1.030	.14	109.01	558.97	**449.96**
125	585	307.2	292.5	0.957	1.046	34.20	110.25	558.98	**448.73**

All parts of machinery which come into contact with ammonia should be constructed of cast iron, wrought iron or steel.

Inflammability of Ammonia Gas.—Pure ammonia is not inflammable and will even extinguish a flame. Where explosions have occurred on account of ammonia coming into contact with a flame, they were due to impurities in the ammonia, mainly oil vapors, which are generated when the ammonia is compressed very hot and when the compressor is lubricated with oils that vaporize at a comparatively low temperature. Such oils are employed because they possess a low congealing point, and it is rather difficult to obtain an oil which has a high evaporating point and at the same time low congealing temperature.

If there is a considerable quantity of oil in a refrigerating system and a mishap occurs, causing a great part of the charge to escape, the oil may be carried along by the escaping ammonia in a finely divided state, and if this should come into contact with a flame, an explosion is liable to occur. If, however, the system is kept clean, an explosion on account of ammonia escaping will not take place, unless a fire, which is sufficiently hot to decompose some of the ammonia into its elements, is present, whence violent explosions at short intervals will occur.

Testing Ammonia.*

Testing for Water by Evaporation.—Screw into the ammonia flask a piece of bent one-quarter inch pipe, which will allow a small bottle to be placed so as to receive the discharge from it. Fill the bottle about one-third full, and throw sample out in order to purge valve, pipe and bottle. Quickly wipe off the moisture which has accumulated on the pipe, replace the bottle and open the valve gently, filling the bottle about half full. This last operation should not occupy more than one minute. Remove the bottle at once and insert in its neck a stopper with a vent hole for the escape of the gas. Procure a piece of solid iron that should weigh not less than 8 or 10 pounds, pour a little water on this and place the bottle on the wet place. The ammonia will at once begin to boil, and in warm weather will soon evaporate. If it shows any residuum, pour it out gently, counting the drops carefully. Eighteen drops are equal to about one cubic centimeter, and if the sample taken amounted to one hundred cubic

* Mostly from Frick Company catalogue.

centimeters, the percentage of liquid remaining can readily be approximated.

Test for Inflammable Gases.—Take a pail of water, submerge the bent end of a quarter-inch pipe therein, open the valve on flask slightly, and allow a small quantity of gas to flow into the water. If inflammable gases are present, they will rise in bubbles to the surface of the water, and may be proved by igniting the bubbles by means of a lighted candle or match. As water has a strong affinity for ammonia, it will readily be absorbed, while air or other gases will show only in the form of bubbles.

Test for Specific Gravity.—The specific gravities of liquid ammonia by Beaumé scale are given in the table page 45. By drawing off some of the liquid in a tall test tube, the Beaumé hydrometer (light) may be inserted and the specific gravity read upon the scale. If water is present, the liquid will show a density proportionate to the percentage of the water present. The specific gravity of pure anhydrous ammonia is 0.623.

Test for Boiling Point.—By inserting a special low temperature standardized chemical thermometer into liquid drawn into the 8-oz. test jar, readings can be obtained through the side of the jar without removing the instrument. Hold the thermometer in such position that only the bulb is immersed. This test will give the boiling point of ammonia at atmospheric pressure, and it is well to know that the state of the barometer affects the temperature of boiling point. With the barometer at 30 inches, the boiling point is about 26.7 degrees below zero. If the ammonia is impure, the boiling point is raised in proportion.

To Test Brine or Water for Ammonia.—"Nessler's Reagent" is used extensively. It is prepared as follows: Dissolve 17 grams of mercuric chloride in about 300 cubic centimeters of distilled water; dissolve 35 grams of potassium iodide in 100 cubic centimeters of water; stir the latter solution into the first until a permanent red precipitate is thrown down. Then dissolve 120 grams of potassium hydrate in 200 cubic centimeters of water and allow the solution to cool, then add to the other solution, and add sufficient water to make one litre. Then add mercuric chloride

solution until a permanent precipitate forms. Let this settle and decant off the clear solution.

Keep the solution in glass stoppered blue bottles. A few drops of this solution added to a sample of brine or water will cause the brine or water to turn yellow if a small percentage of ammonia is present and turn to a full brown if the percentage of ammonia is large.

Impurities Test.—When testing ammonia for impurities, it is diluted with twice its volume of distilled water. It is then made acid with hydrochloric acid. Then to detect the presence of sulphates, add a solution of chloride of barium. If sulphates are present, a white precipitate will be formed. To detect the presence of chlorides the solution of ammonia and water is acidulated with nitric acid instead of hydrochloric, and the white precipitate is formed by the addition of a nitrate of silver solution. But if, in this case, red appears, there is evidence of organic matter.

Refrigerating Effect of Ammonia.

The refrigerating effect of ammonia may be calculated either per pound, or per cubic foot of the vapor. By refrigerating effect is meant the number of B. T. U. which this amount of ammonia is able to abstract from the room to be cooled. On page 497 is given a table of refrigerating effects per cubic foot of ammonia under various conditions.

The refrigerating effect depends on the pressure with which the liquid ammonia reaches the expansion valve (condenser pressure) and its pressure after having passed the expansion valve (suction pressure), and since the pressure of saturated ammonia vapor always corresponds to its temperature, the refrigerating effect of a *pound* of ammonia can be found according to the following formula:

$$h = H - (t_1 - t) \times s.$$

h = refrigerating effect of one pound of ammonia.

H = latent heat of vaporization in B. T. U. of one pound of liquid ammonia at the temperature t.

t_1 = temperature of the liquid in degrees F. before passing the expansion valve.

t = temperature in degrees F. in the expansion coils

Refrigerating Effect in B. T. U. of One Cubic Foot of Ammonia Vapor Compressed and Expanded Between Different Suction and Condenser Pressures.*

Temperature of Vapor in Degrees F.	Corresponding Suction Pressure (Gauge) in Lbs. per Square Inch.	Temperature of Liquid in Degrees F.								
		65	70	75	80	85	90	95	100	105
		Corresponding Condenser Pressure (Gauge) in Lbs. per Sq. In.								
		102.9	114.0	125.9	138.5	152.0	166.4	181.7	198.0	214.8
—25	0.7	28.37	28.03	27.69	27.35	27.01	26.66	26.31	25.96	25.61
—20	3.2	32.65	32.26	31.87	31.47	31.08	30.68	30.28	29.88	29.48
—15	5.8	37.25	36.81	36.37	35.92	35.47	35.02	34.57	34.11	33.65
—10	8.7	42.29	41.79	41.29	40.78	40.27	39.76	39.25	38.73	38.22
— 5	11.8	47.83	47.26	46.70	46.13	45.56	44.98	44.40	43.82	43.24
0	15.3	53.85	53.22	52.58	51.94	51.30	50.66	50.01	49.36	48.70
5	19.2	60.52	59.81	59.10	58.38	57.66	56.94	56.21	55.48	54.75
10	23.3	67.82	67.03	66.24	65.44	64.63	63.83	63.02	62.20	61.38
15	27.9	75.82	74.94	74.05	73.16	72.27	71.37	70.46	69.55	68.64
20	33.0	84.51	83.53	82.55	81.56	80.56	79.56	78.56	77.55	76.54
25	38.5	94.05	92.96	91.87	90.77	89.67	88.56	87.45	86.33	85.21
30	44.5	104.45	103.24	102.04	100.82	99.60	98.37	97.14	95.90	94.65
35	51.0	115.80	114.47	113.13	111.79	110.44	109.08	107.72	106.34	104.97

* In calculating this table, the latent heat and weight of ammonia have been taken from Macintire's Table. The specific heat of liquid ammonia has been calculated in each instance by Dieterici's formula, using the average specific heat between suction and condenser temperatures.

s = average specific heat of the liquid ammonia before and after passing the expansion valve.

The specific heat of liquid ammonia may be calculated according to Dieterici's formula (page 487), or for practical purposes may be taken as 1.1. The values H, t_1 and t can be taken from the ammonia table (page 490).

Example.—What is the refrigerating effect of one pound of ammonia having a pressure of 165 pounds gauge before, and 25 pounds gauge after, passing the expansion valve?

Solution.—By referring to the ammonia table we find that the temperatures t and t_1 corresponding to the pressures 25 and 165 pounds, are 11.9° and 89.5° F. respectively, and that the latent heat of evaporation at the temperature t (11.9° F.) is 560.9. Consequently we have:

$$h = 560.9 - (89.5 - 11.9) \times 1.1 = 475.5.$$

The refrigerating effect is 475.5 B. T. U.

The refrigerating effect of a *cubic foot* of saturated ammonia vapor is of special interest for the designing of the compressor, and for comparison between ammonia and other refrigerating substances. It is found by multiplying the refrigerating effect of one pound of ammonia by the weight of one cubic foot. The formula is therefore the following:

$$h_1 = [H - (t_1 - t) \times s] \times w.$$

h_1 = refrigerating effect per cubic foot.

w = weight in pounds of one cubic foot of ammonia vapor at the temperature t.

Example.—What is the refrigerating effect of one cubic foot of saturated ammonia vapor at a suction pressure of 25 pounds gauge, when it is compressed and condensed to liquid at a pressure of 165 pounds gauge, and then again evaporated at the above suction pressure?

Solution.—The temperatures t_1 and t, corresponding to 25 and 165 pounds pressure are 11.9 and 89.5° F. respectively, the latent heat vaporization at 11.9° F. is 560.9 B. T. U., and the weight of one cubic foot of saturated ammonia vapor at 25 pounds gauge pressure is 0.1412 pound. All these data are taken from the ammonia table (page 490). The refrigerating effect per cubic foot is therefore:

$$h_1 = [560.9 - (89.5 - 11.9) \times 1.1] \times 0.1412 = 67.1 \text{ B. T. U.}$$

The refrigerating effect of one cubic foot of saturated ammonia vapor may also be found by dividing the refrigerating effect of one pound of ammonia by the volume in cubic feet of one pound of ammonia at the suction pressure. As 7.080 cubic feet is the volume of one pound of saturated ammonia vapor at 25 pounds gauge pressure, we obtain, using the figures of the above example:

$$h_1 = \frac{560.9 - (89.5 - 11.9) \times 1.1}{7.080} = 67.1 \text{ B. T. U.}$$

The table on properties of saturated ammonia gives both the weight in pounds per cubic foot, and the volume in cubic feet per pound of saturated ammonia vapor.

The refrigerating effect of other vapors may be found in a similar manner as that of ammonia.

The Compressor.

The following remarks and calculations refer not only to the ammonia compressor, but also to compressors for other refrigerating substances.

The theoretical maximum capacity of a compression machine is dependent on the substance used, and the difference between the theoretical and the actual capacity can, therefore, only be due to losses sustained during the cycle of operations.

Volume of Compressor.

The volume of the compressor is the space through which its piston travels during each stroke. It is expressed by the following formula:

$$v = d^2 \times 0.7854 \times l.$$

v = volume of compressor in cubic feet.
d = diameter of compressor in feet.
l = length of stroke in feet.

If the diameter and the length are given in inches the formula becomes:

$$v = \frac{d^2 \times 0.7854 \times l}{1{,}728}$$

Piston Displacement.

The apparent piston displacement per minute of a double-acting compressor is found by the following formula:

$$C = \frac{d^2 \times 0.7854 \times l \times 2 \times r}{1{,}728}$$

C = theoretical displacement in cubic feet per minute.
d = diameter of compressor in inches.
l = length of stroke in inches.
r = revolutions per minute.

For single-acting machines, the formula also holds good, because a single-acting machine always consists of two single-acting compressors.

If the compressor is to produce a certain amount of actual refrigeration in a stated time, the theoretically required *piston displacement* in cubic feet per minute, can be found by the following formula:

$$V = \frac{Q \times v}{h}$$

V = displacement in cubic feet per minute.
Q = required refrigerating capacity in B. T. U. per minute.
v = volume of one pound of ammonia vapor in cubic feet at suction pressure.
h = refrigerating effect of one pound of ammonia in B. T. U. evaporated from liquid at a given condenser pressure to the suction pressure.

The *actual piston displacement* required is higher, and if we assume the various losses in the entire cycle to be 25%, it may be found by dividing the theoretical displacement by 0.75.

Example.—A compressor should abstract 9.860 B. T. U. per hour from the refrigerating room at a suction pressure of 25 pounds gauge, compressing the ammonia to a pressure of 165 pounds gauge. What must be the piston displacement of the compressor in cubic feet per minute?

Solution.—The refrigerating effect per pound of ammonia at the pressure given in the example, is 475.5 B. T. U. (see page 498), and the volume of one pound at the suction pressure is 7.080 cubic feet. Hence theoretically the *piston displacement* in cubic feet per minute would have to be:

$$V = \frac{9{,}860 \times 7.080}{475.5 \times 60} = 2.447 \text{ cubic feet.}$$

If the losses are taken at 25%, then the *actual piston displacement* would have to be:

$$V = \frac{2.447}{0.75} = 3.263 \text{ cubic feet.}$$

This agrees very closely with results found in actual practice, as will be shown later.

The piston displacement per minute varies with the nature of the fluid used for refrigerating. For ammonia, manufacturers generally allow a capacity of 7,200 to 7,500 cubic inches per ton, though the figure may vary considerably for different cases. The following table gives the approximate average piston displacement per minute per ton refrigeration in 24 hours for ammonia, carbon dioxide, and sulphur dioxide:

Substance.	*Piston displacement per minute.*
Carbon dioxide	1,300 cubic inches.
Ammonia	7,200 cubic inches.
Sulphur dioxide	18,900 cubic inches.

From this it can be seen that for producing the same amount of refrigeration, a sulphur dioxide compressor must handle 2.625 times as much gas as an ammonia compressor, and the latter 5.5 times as much as a carbon dioxide compressor.

For ammonia, 7,200 cubic inches of piston displacement is sufficient, when back pressures of 20 to 27 pounds are carried, but for 15 pounds back pressure, 7,500 cubic inches is more correct. The piston displacements for carbon dioxide and sulphur dioxide vary in the same proportion, with varying conditions.

Volumetric Efficiency.

The term volumetric efficiency may refer either to the apparent volumetric efficiency or the actual volumetric efficiency. By *apparent volumetric efficiency,* as commonly understood, is meant the percentage of gas discharged per stroke as compared to the amount of gas which would be discharged were there no re-expansion. This is determined from the indicator card, i. e., the apparent volumetric efficiency is the ratio between the distance from the compression end of the diagram to the point where the re-expansion line meets the suction line extended, and the total length of the diagram. It is usually expressed in per cent, so that the above ratio must be multiplied by 100.

This figure does not take into account suction depression through the suction valves and expansion pipe system, nor does it consider the superheating of the vapor in the cylinder. When

these items are considered, in addition to the re-expansion, we have what is termed *actual volumetric efficiency.*

It is reasonable that the lowering of the suction pressure (suction depression) between a point just beyond the suction valve and the cylinder should be charged to the compressor. This loss will naturally be greater with small valve areas than with large valve areas. The loss due to heating of the gas in the cylinder must also be considered in calculating the actual volumetric efficiency. The cold gas coming in contact with the hot cylinder walls, valves and piston, is heated, and in such condition weighs less per unit of volume. Consequently, the cylinder does not hold as much gas by weight as that calculated from the card. Therefore, in determining the actual volumetric efficiency, the lowering of the suction pressure through the suction valve and the superheating are also considered in addition to the clearance loss. An approximate value of actual volumetric efficiency could be arrived at without considering the loss due to superheating, by determining the suction pressure in the line just before the gas enters the suction valve. The approximate value would then be the apparent volumetric efficiency as determined, times the absolute suction pressure, as read on the indicator card, divided by the absolute suction pressure as determined in the suction line near the suction valve.

To arrive accurately at the actual volumetric efficiency however, involves the determination of the weight of ammonia passing through the system in a certain time, calculating the cooling effect of this and dividing by the cooling effect calculated from the amount of ammonia as determined from the apparent piston displacement, that is, the volume of ammonia which would be discharged were there no loss of any kind in the cylinder.

Example.—The length of the indicator diagram of a 12x22 double-acting compressor is 3¾ inches. The distance from the compression end of the card to the point where the re-expansion line meets the suction line extended is $3^{9}/_{16}$ inches. What is the apparent volumetric efficiency?

Solution.—The apparent volumetric efficiency is:

$$\frac{3^{9}/_{16} \times 100}{3\frac{3}{4}} = \frac{57 \times 4 \times 100}{16 \times 15} = 95\%.$$

The per cent of re-expansion in the above case is 5%.

Capacity of Compressor in Tons Refrigeration.

The formula for the *theoretical capacity*, in tons of refrigeration per 24 hours, of a double-acting compressor, taking into consideration the specified pressures in the condenser and the refrigerator, is the following:

$$T = \frac{d^2 \times 0.7854 \times l \times 2 \times r \times h \times 1{,}440}{1{,}728 \times 288{,}000}$$

T = tons refrigeration in 24 hours.
d = diameter of compressor in inches.
l = stroke in inches.
r = number of revolutions per minute.
h = refrigerating effect of one cubic foot of ammonia vapor at refrigerator pressure, completely evaporated from liquid at condenser pressure.
1,440 = minutes in 24 hours.
288,000 = B. T. U. equivalent to one ton refrigeration.

The formula holds good for any refrigerating substance; the only item to be changed being the value of h, which varies with the different substances. The same formula is also used for single-acting compressors with two cylinders.

Example.—A double-acting Linde compressor of 9⅞ inches diameter and 16⅞ inches stroke, making 70 revolutions per minute, takes the ammonia in at 25 pounds gauge pressure and discharges it at 165 pounds gauge pressure. What is its capacity in tons of refrigeration in 24 hours?

Solution.—The refrigerating effect of one cubic foot of ammonia vapor expanded between the pressures given in the example is 67.1 (see page 499); consequently we have:

$$T = \frac{9\frac{7}{8} \times 9\frac{7}{8} \times 0.7854 \times 16\frac{7}{8} \times 2 \times 70 \times 67.1 \times 1.440}{1{,}728 \times 288{,}000} = 35.42$$

tons refrigeration in 24 hours.

This is the *theoretical capacity* of the compressor. The *actual capacity* is smaller, since the various losses amount, on the average, to about 25%. In the above example, where the dimensions given are those of an ordinary 25-ton Linde machine, the actual capacity under the condenser and refrigerator conditions specified, would therefore be

$$35.42 \times 0.75 = 26.56 \text{ tons of refrigeration.}$$

It has been stated above that a capacity of 7,200 cubic in-

ches per minute is generally allowed for ammonia compressors per ton of refrigeration in 24 hours. Based on this, the formula for calculating the capacity in tons may be expressed as follows:

$$T = \frac{d^2 \times 0.7854 \times l \times 2 \times r}{7{,}200}$$

T = tons of refrigeration in 24 hours.
d = diameter of compressor in inches.
l = length of stroke in inches.
r = revolutions per minute.

Example.—Assuming the same compressor and conditions as above, and allowing 7,200 cubic inches displacement per ton, what is the capacity of the machine in tons?

Solution.—

$$T = \frac{9\frac{7}{8} \times 9\frac{7}{8} \times 0.7854 \times 16\frac{7}{8} \times 2 \times 70}{7{,}200} = 25.1.$$

The capacity of the machine is approximately 25 tons.

The capacity of a compressor varies at different pressures, the same compressor with the same number of revolutions having a greater capacity, the nearer the suction and condenser pressures are to each other. This can easily be proven by the following examples:

Example 1.—A double-acting compressor, 10 inches in diameter, having an 18-inch stroke and making 65 revolutions per minute, takes in ammonia at 33 pounds gauge pressure and discharges it against 150 pounds gauge pressure. What is the capacity of the compressor?

Solution.—The refrigerating effect of one cubic foot of saturated ammonia vapor expanded between the pressures given, is found from the formula on page 498, as follows:

$$h = \frac{554.47 - (84.3 - 20) \times 1.1}{5.951}$$

$= 81.29$, or approximately 81.3.

Substituting this value in the formula for tons of refrigeration given on page 503, we obtain:

$$T = \frac{10 \times 10 \times 0.7854 \times 18 \times 2 \times 65 \times 81.3 \times 1{,}440 \times 0.75}{1{,}728 \times 288{,}000}$$

$= 32.42$ tons.

Example 2.—A compressor of the same dimensions as that in the preceding example takes the ammonia in at 25 pounds

gauge pressure and discharges it at 165 pounds. What is the capacity of the compressor under these conditions?

Solution.—The refrigerating effect of one cubic foot of saturated ammonia vapor evaporated between the pressures given is 67.1 (see page 498), and we consequently have:

$$T = \frac{10 \times 10 \times 0.7854 \times 18 \times 2 \times 65 \times 67.1 \times 1{,}440 \times 0.75}{1{,}728 \times 288{,}000}$$

$= 26.76$ tons.

In the first example, where the difference between the two pressures is only 117 pounds, the capacity is 32.42 tons. In the second case, the difference in pressures is 140 pounds, and the capacity is thereby reduced to 26.76 tons of refrigeration.

Diameter of Compressor.

The diameter of a compressor necessary for a certain refrigerating capacity can be calculated by the following formula, which is obtained from the formula on page 503, expressing the refrigeration in B. T. U., and using the refrigerating effect per pound.

$$d = \sqrt{\frac{Q \times v \times 1{,}728}{h \times l \times r \times 2 \times 0.7854}}$$

d = diameter of the compressor in inches.
Q = required refrigerating capacity in B. T. U. per minute.
v = volume of one pound of saturated ammonia vapor in cubic feet at suction pressure.
h = refrigerating effect of one pound of ammonia.
l = length of stroke in inches.
r = number of revolutions per minute.

Example.—What must be the diameter in inches of a double-acting compressor supposed to produce 50 tons of refrigeration in 24 hours, if the compressor takes in the ammonia at 25 pounds suction pressure and discharges it at 165 pounds condenser pressure? The length of stroke is to be 22 inches and the revolutions per minute are to be 60. The various losses during the cycle amount to 25%.

Solution.—Since the compressor is supposed to produce 50 tons of refrigeration in 24 hours, its refrigerating capacity, expressed in B. T. U. per minute, is $\frac{50 \times 288{,}000}{1{,}440} = 10{,}000$ (the figure 1,440 is the number of minutes in 24 hours). The refrig-

erating effect h of one pound of ammonia, is found according to the formula given previously, and amounts to 475.5 B. T. U. The volume of one pound of ammonia vapor at 25 pounds suction pressure is found from the ammonia table to be 7.08 cubic feet. As the losses amount to 25%, we divide by 1.00—0.25, or 0.75, as indicated.

Substituting all these values we obtain as the diameter of the compressor:

$$d = \sqrt{\frac{50 \times 288{,}000 \times 7.08 \times 1{,}728}{1{,}440 \times 475.5 \times 22 \times 60 \times 2 \times 0.7854 \times 0.75}} = 12.9.$$

The diameter consequently should be 13 inches.

The diameter can also be calculated approximately according to the following formula, obtained by transposing the empirical formula given on page 504:

$$d = \sqrt{\frac{T \times 7{,}200}{r \times l \times 2 \times 0.7854}}$$

d = diameter of cylinder in inches.
T = tons of refrigeration in 24 hours.
r = number of revolutions per minute.
l = length of stroke in inches.

Example.—What must be the diameter of a 50-ton double-acting compressor, stroke to be 22 inches and the machine to make 60 revolutions per minute?

Solution.—

$$d = \sqrt{\frac{50 \times 7{,}200}{60 \times 22 \times 2 \times 0.7854}} = 13.2 \text{ inches.}$$

Quantity of Ammonia Circulated.

The number of pounds of ammonia to be circulated per hour in order to obtain a certain amount of refrigeration, can be found by dividing the B. T. U. of refrigeration required per hour, by the refrigerating effect of one pound of ammonia under existing conditions, or expressed in a formula:

$$W = \frac{288{,}000 \times T}{24 \times h}$$

W = pounds of ammonia circulated per hour.
T = tons of refrigeration in 24 hours.
h = refrigerating effect of one pound of ammonia.

Example.—How many pounds of ammonia must be circulated per hour for a 50-ton refrigerating machine, if the suc-

tion pressure is 25 pounds and the condenser pressure 170 pounds gauge?

Solution.—The refrigerating effect of one pound of ammonia under these conditions is

$$560.9 - (91.2 - 11.9) \times 1.1 = 473.7.$$

Therefore:

$$W = \frac{288{,}000 \times 50}{24 \times 473.7} = 1{,}267.$$

There are 1,267 pounds of ammonia required per hour or $\frac{1{,}267}{60} = 21.1$ pounds per minute, or $\frac{21.1}{50} = 0.42$ pound per minute per ton.

In practice, however, there is a loss of about 12% on account of radiation, refrigeration loss in cooling the liquid, compression heat, etc., which brings the quantity up to $\frac{21.1}{0.88} =$ approximately 24 pounds of ammonia per minute for 50 tons refrigeration in 24 hours. Since a cubic foot of ammonia at 25 pounds gauge pressure weighs 0.1412 pound, $\frac{24}{0.1412} = 170$ cubic feet must be discharged by the compressor per minute. The piston displacement must be larger on account of piston and valve leakage, and re-expansion of ammonia which remained in the clearance space, the loss being about 13%. Based on this figure, the displacement would therefore have to be $\frac{170}{0.87} =$ approximately 195 cubic feet.

Usually the amount of ammonia required for a refrigerating system is determined according to an empirical rule, and the calculation is based on the length of the expansion coils, 2-inch pipes being taken as standard. For every lineal foot of 2-inch pipe, ¼ pound of ammonia is required in the system, for smaller pipes correspondingly less. Thus, a refrigerating plant containing 10,000 lineal feet of 2-inch pipe must receive a charge of 2,500 pounds of ammonia, or approximately 25 drums of 100 pounds each. To this, however, an addition must be made for the condenser, liquid receiver, oil traps, etc., as follows:

For a 10-ton machine..... 200 pounds of ammonia.
For a 15-ton machine..... 250 pounds of ammonia.
For a 25-ton machine..... 350 pounds of ammonia.

For a 35-ton machine..... 400 pounds of ammonia.
For a 50-ton machine..... 450 pounds of ammonia.
For a 65-ton machine..... 500 pounds of ammonia.
For a 100-ton machine..... 580 pounds of ammonia.
For a 150-ton machine..... 680 pounds of ammonia.
For a 200-ton machine..... 780 pounds of ammonia.
For a 300-ton machine.....1,080 pounds of ammonia.
For a 400-ton machine.....1,380 pounds of ammonia.

Assuming, for instance, that the system of a 65-ton refrigerating plant is equipped with 14,000 lineal feet of 2-inch pipe, the total charge of ammonia would then amount to $\frac{14,000}{4} + 500$, or 4,000 pounds, which corresponds to approximately 40 drums of 100 pounds each.

Influence of Clearance in Compressor.

The influence of clearance is not so detrimental to the *efficiency* of a compressor as has been claimed by a number of writers on refrigeration. It is, however, very detrimental to the *capacity* of the compressor, and the more so, the greater the difference between the two pressures.

From a theoretical standpoint, clearance has no influence on the work required to compress the gas, since the work performed on the gas remaining in the clearance space is again given off to the piston during the return stroke. The condition of the enclosed fluid is the same at the beginning and end of the expansion as it was at the end and beginning of the compression. In practice, however, this is not true, because the expansion of the compressed gas is not adiabatic, as it is in contact with a large surface through which heat is radiated or conducted. This means that when the gas is in the compressed state it is hotter than the surrounding air, and heat is consequently radiated into the air and stored up in the cylinder walls, piston, valves, etc., whereas, when the gas first enters the compressor from the refrigerator, it is colder than the surrounding air, and at this stage, heat is conducted into the ammonia gas from the air, cylinder walls, piston, valves, etc., thereby increasing its volume. In the former case, the radiation of heat will cause a loss of work, while in the latter case, capacity is lost in the same ratio as heat is conducted into the ammonia.

The friction in a compressor is practically the same whether all or none of the gas is discharged, and therefore, the percentage loss due to friction will increase for a certain refrigerating effect when the capacity of the compressor decreases.

For keeping the capacity uniform, the clearance should be as small as possible, and should not exceed 7% of the compressor volume. Some makers build their compressors so that the piston strikes each head if the latter is bolted on without a gasket, and the thickness of the gasket is supposed to be the clearance.

When a compressor is in operation, the slack in the bearings furnishes additional clearance. This is liable to be more than is generally supposed, as there are three bearings that can produce excessive clearance, viz., the pillow-block, the crank-pin and the cross-head bearings. With loose bearings the compressor will naturally not run smoothly, and it is therefore always a sign of excessive clearance when a compressor is *knocking.*

The exact amount of re-expansion is best determined by the *compressor indicator diagram,* where it is represented by the distance from the end of the diagram to the point where the expansion line meets the suction pressure line extended (see also pages 502 and 562).

In a double-acting compressor, the clearance represents a fixed amount and cannot be less than what it is built for. In a single-acting compressor, however, the clearance can be reduced to the lowest practical minimum, and in this respect, the single-acting compressor offers an advantage over the double-acting.

In some ammonia and carbon dioxide refrigerating machines, the attempt was made to reduce the clearance space by filling it up with sealing oil, so that the piston dips into the oil at the end of the stroke. This method, irrespective of how correct it may appear in theory, does not reduce the re-expansion any more than it amounts to in machines that were built like those described above. The sealing oil becomes foamy, and to a large extent absorbs the high tension ammonia vapors, giving them off again at low tension during the return stroke, the oil

thus producing the same effect as the clearance does in other well-built machines.

As this system has other disadvantages, serious trouble may arise if it is placed into ignorant hands. The system is therefore being gradually discarded.

Liquid in Clearance Space.—The description of clearance just given refers to clearance space filled with gas only. If, however, liquid, or a mixture of gas and liquid is entrapped in the clearance space, the economy as well as the capacity is seriously affected. The re-expansion in a compressor will be increased considerably by the expanding liquid, but since very little work is done by this expanding liquid, the loss is increased. It is therefore a mistake to operate any machine very wet, i. e., to admit much liquid into the compressor. More will be said about this under compressor diagrams (see page 553).

Friction in Compressors.

Friction in compressors represents a loss of the work produced by the steam engine. It varies in amount with the construction of the refrigerating machine. Its relative effect is smaller in carbon dioxide than in ammonia or sulphur dioxide machines, greater in ether machines, and relatively the greatest in air machines.

Not only does friction in the compressor represent a direct loss of work, but it also generates heat, which is given off to the working fluid. The compression of the fluid is not exactly adiabatic, because heat is absorbed by conduction when the gas enters the compressor, and still more is lost during compression, by radiation from the quantity remaining in the compressor.

The range between the initial and final temperatures in the compressor is very wide, and in machines that work without superheating, the average initial temperature is below the freezing point, so that it often happens that the inner cylinder walls are covered with a dew.

The heat added to compression from the various causes, amounts, in the case of ammonia machines, to from 1 to 2 per cent of the refrigeration produced.

Filling Losses of Compressors.

If the compressor is not properly filled with gas at the same pressure as exists directly beyond the expansion valve, this must

be considered a loss. A reduction in pressure is mostly caused by friction of the gas in the expansion coils, and when entering through the suction valves. It not only increases the work that must be furnished for compression, but on account of the volume of the gas being increased by the decrease in pressure, also lowers the refrigerating capacity. This loss is the greater the more volatile the fluid is, and is therefore the greatest in carbon dioxide machines.

The loss caused by filling the compressor with gas of a lower pressure than is created by the expansion valve is called *suction depression,* and on the average amounts to about 5%. For instance, if the suction pressure a short distance beyond the expansion valve is 25 pounds, the indicator diagram from the compressor will show about $25 - \frac{25 \times 5}{100}$ or 23.75 pounds of pressure.

Compound Compressors.

Where there is great difference in temperatures, and therefore also in pressures, in the refrigerating system, for instance, when low temperatures must be kept in the refrigerator and when the temperature of the cooling water is high, compound compressors may be used to advantage. These compressors perform compression in two stages. Clearance, cylinder heating, leakage, etc., are considerably reduced, and the capacity and economy of the machine are therefore increased correspondingly. They are especially well adapted for use in southern climates where the temperature of the condenser water is generally high.

Compound compressors of large size and connected with their crank shafts at right angles, insure a much more uniform motion than ordinary compressors, since the load is more uniformly distributed. The friction, however, is greater than if the compressors were connected in tandem.

Multiple Effect Compressor.

The multiple effect compressor patented by G. T. Voorhees overcomes the disadvantage occasioned by the fact that both the economy and the capacity of an ordinary compression machine vary with the back pressure. Instead of only running the compressor at a suction pressure corresponding to the lowest required temperature, this device takes in gas from two or more refriger-

ators at two or more different suction pressures in the same cylinder. At the start and during the full suction stroke, the cylinder receives low pressure gas. At the end of the suction stroke, the cylinder thus being first filled with low pressure gas, high pressure gas enters and compresses this low pressure gas up to its pressure and thus makes room for itself. Any compressor can be made multiple effect by placing a multiple effect compressor device in the suction pipe.*

Flooded System.

This system consists in filling the entire refrigerating pipe coils with liquid ammonia. It is sometimes called the gravity feed system, which name is really better suited to it.

In addition to the expansion coils, a so-called accumulator is attached above the expansion coils, the ammonia being fed into this accumulator. The bottom outlet enters the bottom header of the expansion coils, and the top is connected to about the middle of the accumulator. The ammonia thus flows into the coils by gravity. The ammonia gas is liberated at the surface of the accumulator and passes from there to the compressor.

The best kind of flooded system is that in which the ammonia enters through a coil inserted in the accumulator, whereby the liquid ammonia is cooled to the temperature of the boiling liquid before it flows into the accumulator, so that the ammonia flows practically like water.

The advantages of a flooded system as compared to a force feed system are as follows: The entire cooling surface of the refrigerating pipes is made very effective, that is, it transmits the maximum amount of heat under the prevailing conditions. It furthermore does away with the numerous expansion valves, the entire feed being controlled by one inlet valve. There are practically no disadvantages when this system is used for ice freezing tanks, ordinary brine tanks, double-pipe coolers and Baudelot coolers, but it is not very well adapted for refrigerating rooms, as such rooms are usually placed one above the other, nor is it well adapted where different temperatures should be carried.

With the flooded system, wherever it can be used, the same

* See "Ice and Refrigeration," Vol. 32, 1907, P. 52.

effect can be obtained with about 20% less pipe surface than that necessary for the ordinary force feed system. It does not, however, increase the capacity of the refrigerating machine, the gain being in making the refrigerating surface more effective.

Work Required on Compressor.

The theoretical work necessary to produce a certain amount of refrigeration can be calculated in B. T. U. according to this formula:

$$W = \frac{h \times (t_1 - t)}{t + 460}$$

W = theoretical work done in B. T. U.
h = refrigerating effect of one pound of ammonia in B. T. U.
t_1 = temperature in condenser in degrees F.
t = temperature in refrigerator in degrees F.

Example.—What amount of work in B. T. U. is theoretically necessary in the compressor for the refrigeration produced by one pound of ammonia, the gas entering the compressor at 20 pounds gauge pressure and leaving it at 170 pounds gauge pressure?

Solution.—The temperatures corresponding to these pressures are 91.2 and 6° F. respectively. The refrigeration produced by one pound of ammonia under these conditions can be found according to the formula given on page 498.

$$h = 565.4 - (91.2 - 6) \times 1.1 = 471.7 \text{ B. T. U.}$$

Since the refrigeration amounts to 471.7 B. T. U., the work necessary to produce this refrigeration must be:

$$W = \frac{471.7 \times (91.2 - 6)}{6 + 460} = 86.2 \text{ B. T. U.}$$

To produce 471.7 B. T. U. of refrigeration, under the conditions given, a theoretical compressor work is required which corresponds to 86.2 B. T. U. or $86.2 \times 778 = 67{,}064$ foot-pounds.

Theoretical Horse-Power Required per Ton of Refrigeration.

The number of horse-power theoretically necessary to produce one ton of refrigeration per 24 hours can be calculated in a similar manner:

$$H.P. = \frac{288{,}000 \times 778 \times (t_1 - t)}{33{,}000 \times 1{,}440 \times (t + 460)}$$

t_1 = temperature in condenser in degrees F.

t = temperature in refrigerator in degrees F.
288,000 = number of B. T. U. equal to one ton of refrigeration.
778 = number of foot-pounds equivalent to one B. T. U.
33,000 = number of foot-pounds per minute equal to one horse-power.
1,440 = number of minutes in 24 hours.

Example.—How many horse-power are theoretically required for the production of one ton of refrigeration per 24 hours, if the ammonia gas enters the compressor at 20 pounds gauge pressure and leaves it at 170 pounds gauge pressure?

Solution.—The temperatures corresponding to the pressures are 91.2° F. and 6° F. respectively. Therefore, we have:

$$\text{H. P.} = \frac{288{,}000 \times 778 \times (91.2 - 6)}{33{,}000 \times 1{,}440 \times (6 + 460)} = 0.86.$$

Theoretically 0.86 H. P. are necessary.

With different condenser and refrigerator pressures, the horse-power required varies for the same amount of refrigeration.

The smaller the difference between condenser and suction pressures, and consequently between the temperature of the gas entering and leaving the compressor, the less horse-power will be required. This can be illustrated by comparing the result of the following example with that of the preceding one:

Example.—The suction pressure is 30 pounds gauge, the condenser pressure 140 pounds gauge. How many horse-power are theoretically necessary for the production of one ton of refrigeration in 24 hours?

Solution.—The temperatures corresponding to the pressures given are 17 and 81° F. respectively. Consequently:

$$\text{H. P.} = \frac{288{,}000 \times 778 \times (81 - 17)}{33{,}000 \times 1{,}440 \times (17 + 460)} = 0.63.$$

It will be perceived that in the latter case only 0.63 H. P. is theoretically necessary for the production of one ton of refrigeration, whereas it required 0.86 H. P. under the conditions of the preceding example for exactly the same amount of refrigeration.

In the above examples no losses of any kind were taken into account. The results are interesting, in so far as they show the disadvantage of carrying the suction pressure much lower than is necessary. However, the installing of insufficient pipe

surface in the refrigerated rooms, as well as the use of too small a suction pipe or obstructions in the latter are often the cause of this deplorable condition. The economy of the machine must of course, suffer thereby.

Actual Horse-Power Required per Ton of Refrigeration.

The actual power required is, of course, larger than the theoretical, there being losses due to friction, superheating, etc. The loss in the compressor amounts to from to 12 to 20%.

Example.—How many horse-power are actually required for the compressor of a 50-ton refrigerating machine, working with 25 pounds suction and 170 pounds condenser gauge pressure, assuming a loss in this case of 17% ?

Solution.—H. P. $= \frac{50 \times 288{,}000 \times (91.2 - 12) \times 778}{33{,}000 \times 1{,}440 \times (12 + 460) \times 0.83} = 47.5.$

In this case, 47.5 brake H. P. are required. There is, however, a further loss in the driving mechanism, which in the case of a steam engine amounts on an average to about 25%. The actual indicated horse-power required, under these conditions would be $\frac{47.5}{0.75} = 63.3$ H. P., or $\frac{63.3}{50} = 1.27$ H. P. per ton.

The amount of power actually required averages from 1.25 to 2 H. P. per ton of refrigeration, varying with the size of the machine, and condenser and refrigerator pressures. On the average, a 6-ton machine requires 2 H. P. per ton, a 25-ton 1.54 H. P. per ton, a 50-ton 1.27 H. P. per ton, and a 100-ton 1.25 H. P. per ton of refrigeration.

Under average conditions, about 37 pounds of steam are used in a Corliss engine (non-condensing) per hour for every ton of refrigeration that a 50-ton machine produces in 24 hours.

For every compressor horse-power per hour, about 12,000 B. T. U. can be abstracted from the refrigerating room, and for every engine horse-power per hour, about 9,400 B. T. U. If the engine uses 30 pounds of steam per horse-power per hour, then, for approximately every 3 B. T. U. furnished to the engine, 1 B. T. U. can be abstracted from the refrigerating room.

The above figures have frequently been found in practice by the author.

The *best condition of the compressed ammonia gas* at the

end of compression in order to require the least work, is when it is in the state of a dry, saturated gas. This condition is, however, not easily obtainable in practice, nor is it on the other hand harmful to work the gas in a medium superheated state, as is the case in dry compression machines.

Performance and Efficiency of Compressors.

The actual performance of an ammonia or any other kind of compressor is best determined by taking indicator cards. The ammonia indicator card gives the pressure from and to which the ammonia is compressed, and the amount of re-expansion, or the amount of ammonia which is not discharged. Furthermore, it indicates the exact conditions under which the ammonia is compressed. This is necessary in order to determine the capacity of the compressor. For determining the mechanical efficiency and economy, however, it is also necessary to take indicator cards of the steam engine driving the compressor at the same time as the ammonia cards are taken. The difference in power shown by the two cards, the horse-powers being calculated according to the method given for steam engines on page 329, represents the frictional loss of the machines. For machines of medium size (about 50 tons), the compressor card generally represents about 25% less power than the steam engine card shows; for smaller machines the friction is comparatively larger.

A detailed description and criticism of indicator cards is given later.

The term *efficiency of compressor* may have various meanings according to whether the performance of the compressor is compared with the performance of the engine, the performance of the boiler, the coal consumption, the horse-power expended, the refrigeration produced in proportion to the heat expended, etc. In order to avoid confusion, it is, therefore, always necessary to state what the efficiency refers to.

Actual Efficiency.

By the efficiency of a compressor is generally understood the number of B. T. U. abstracted from the refrigerating room for every B. T. U. of work done by the compressor.

It can be expressed by the formula:

$$E = \frac{H}{W}$$

E = efficiency of the compressor in reference to work done by it.
H = number of foot-pounds corresponding to heat absorbed in B. T. U. by one pound of ammonia in the refrigerator.
W = work expressed in foot-pounds required to raise the temperature of one pound of ammonia from the suction to the discharge temperature.

Example.—The compressor of a 100-ton machine requires on the average 0.95 H. P. at the compressor to compress sufficient ammonia per minute for the production of one ton of refrigeration in 24 hours, from a suction pressure of 25 pounds gauge to a condenser pressure of 168 pounds gauge. What is the efficiency of the compressor?

Solution.—Before calculating the efficiency, the power required to compress one pound of ammonia per minute must first be found.

The refrigerating effect of a pound of ammonia under the above conditions is 473.7 B. T. U. (see page 507).

Assuming a loss of 12%, the power required to compress one pound of ammonia per minute is then:

$$\frac{473.7 \times 60 \times 24 \times 0.95}{288{,}000 \times 0.88} = 2.56 \text{ H. P.}$$

The efficiency can now be calculated according to the formula given above:

$$E = \frac{473.7 \times 778}{2.56 \times 33{,}000} = 4.362.$$

This means that for each B. T. U. of work done by the compressor, 4.362 B. T. U. are abstracted from the refrigerating rooms.

The result calculated according to the above formula furnishes the actual efficiency for a given case, which, of course, varies with the working conditions, viz., with the temperatures in the refrigerator and condenser.

Theoretical Maximum Efficiency.

The theoretical maximum efficiency E_1 is expressed by Linde in the formula:

$$E_1 = \frac{T}{T_1 - T}$$

T = absolute temperature in degrees F. in the refrigerator.
T_1 = absolute temperature in degrees F. in the condenser.

From the foregoing, it can clearly be seen that the refrigeration which can be obtained by expending a certain amount of work is the greater the higher the temperature T is, and the lower the temperature T_1 is, or the smaller the difference between the two temperatures, T_1 and T.

Example.—The ammonia gas enters the compressor at 25 pounds gauge pressure and leaves it at 168 pounds gauge pressure, the gas being in a saturated state. What does the theoretical maximum efficiency amount to under these conditions?

Solution.—The temperatures corresponding to the pressures given are found in the ammonia table to be 11.9° F. and 90.5° F. We have consequently:

$$E_1 = \frac{11.9 + 460}{(90.5 + 460) - (11.9 + 460)} = 6.00.$$

The theoretical efficiency is 6.00, which means that for every B. T. U. of work done by the compressor, theoretically 6 B. T. U. can be abstracted from the refrigerating room.

Amount of Work Lost.

The theoretical efficiency is naturally higher than the actual, and for a proper comparison of different tests made with various machines or plants, the ratio between the actual and the theoretical maximum efficiency must be calculated in each case. This ratio, designated by the letter R, is expressed in the formula:

$$R = \frac{E}{E_1}$$

E = actual efficiency.

E_1 = theoretical maximum efficiency.

Example.—The actual efficiency of an ammonia compressor is 4.362 as found in the example on page 517, and its theoretical maximum efficiency is 6.00. What is the ratio R between the two efficiencies?

Solution.—$R = \frac{4.362}{6} = 0.727.$

Expressed in per cent, the ratio is 72.7%. This means that where theoretically 100 B. T. U. could be abstracted from the refrigerating room, in reality only 72.7 B. T. U. are abstracted, so that 27.3% of the work done by the compressor is lost.

This loss consists of leakage, clearance loss, etc. If the loss in the driving mechanism is taken into account, i. e., the fric-

tion of the same, which with a steam engine, on the average amounts to about 25%, then the efficiency compared with the steam engine becomes $4.362 \times 0.75 = 3.2715$, which means that for every B. T. U. of work done by the steam engine, 3.2715 B. T. U. are abstracted from the refrigerating room. The efficiency of a simple non-condensing Corliss engine in good condition and under favorable circumstances is seldom more than 10%, and the average efficiency of boiler and furnace usually 65%, hence the B. T.U. that can be abstracted from the refrigerating room per B. T. U. furnished by the coal are: $4.362 \times 0.10 \times 0.65 = 0.28353$, or for every B. T. U. abstracted from the refrigerating room, $\frac{1}{0.28353} = 3.53$ B. T. U. must be furnished to the furnace under the conditions named above. Under average conditions, the efficiency is not as high as this, but is nearer 5 B. T. U. per B. T. U. abstracted from the refrigerating room, as shown below.

Maximum Theoretical Efficiency of Plant.

The maximum theoretical efficiency of the entire plant can be calculated in a manner similar to the methods given above for the efficiency of the compressor. In this case, the range of temperature is compared with the heat furnished by the coal.

Assuming the engine and compressor to be ideal and the cycles of operation to be Carnot's cycles, while the furnace possesses an efficiency of 65%, the maximum theoretical efficiency of the entire plant is expressed by the formula:

$$E_1 = \frac{T}{T_1 - T} \times \frac{T_2 - T_3}{T_2} \times 65.$$

E_1 = maximum theoretical efficiency of plant in per cent.
T = absolute temperature in degrees F. in refrigerator.
T_1 = absolute temperature in degrees F. in condenser.
T_2 = absolute temperature in degrees F. of steam leaving boiler.
T_3 = absolute temperature in degrees F. of steam leaving engine.

Example.—The temperature of the steam leaving the boiler is 338° F., when leaving the engine 214° F. The temperature of the ammonia in the refrigerator is 12° F. and on leaving the com-

pressor is 90° F. The efficiency of the boiler and furnace is 65%. What is the maximum theoretical efficiency of the plant?

Solution.—

$$E = \frac{12 + 460}{(90 + 460) - (12 + 460)} \times \frac{(338 + 460) - (214 + 460)}{338 + 460} \times 65 = 61.1\%.$$

This means that for every 100 B. T. U. in the coal as furnished to the boiler, 61.1 B. T. U. could be abstracted from the refrigerating room, if there were no losses caused by friction, clearance, or leakage, and if the cycles were Carnot's.

Actual Efficiency of Plant Based on Heat Abstracted from Refrigerating Room Compared to Heat in Coal.

In reality, the efficiency of a plant is considerably lower than the maximum theoretical efficiency shows, which can clearly be seen from an example:

Example.—For driving the compressor given in the examples under "Efficiency of Compressor," a simple non-condensing Corliss engine, which consumes about 30 pounds of steam per H. P. per hour, is employed. If the friction of engine and compressor is 25%, and the efficiency of boiler and furnace is 65%, what is the efficiency of the plant?

Solution.—It has been previously calculated (page 517) that 2.56 H. P. are required by the compressor for compressing one pound of ammonia under the conditions named. Since there is a loss of 25% due to friction, the actual horse-power necessary to compress one pound of ammonia per minute, having a refrigerating effect of 473.7 B. T. U., is therefore, $\frac{2.56}{0.75} = 3.41$, and the actual efficiency referred to the B. T. U. in the coal is thus:

$$\frac{473.7 \times 60 \times 65}{3.41 \times 30 \times 970.4} = 18.61.$$

This signifies that for every 100 B. T. U. in the coal, only 18.61 were abstracted from the refrigerating room. The loss consequently amounts to 81.39%.

If a compound condensing engine, which consumes 16 pounds of steam per horse-power per hour, is used, then the efficiency of the plant is somewhat higher, viz.:

$$\frac{473.7 \times 60 \times 65}{3.41 \times 16 \times 970.4} = 34.9.$$

In this case 34.9 B. T.U. are abstracted from the refrigerating room for every 100 B. T. U. contained in the coal. From the foregoing it can plainly be seen that the efficiency of the plant is proportional to the steam consumption of the engine.

AMMONIA CONDENSERS.

There are three kinds of condensers commonly in use, known respectively as the *submerged,* the *atmospheric,* and the *double-pipe* condenser.

Submerged Condenser.

The submerged condenser derives its name from the fact that the ammonia coils are submerged in the cooling water. The latter enters the tank at the bottom and leaves it at the top, while the ammonia enters the submerged coils at the top, leaving them at the bottom.

The ammonia which leaves the condenser ought in this manner to be cooled to the lowest temperature the cooling water possesses, but since the latter is in very slow motion, a film of warm water surrounds the pipes and not all the cold water comes in contact with them. There is also a tendency for air bubbles to adhere to the surface of the pipes, thus interfering with the cooling. On account of the large body of water which surrounds the coils, leaks are not readily detected, as the escaping ammonia is at once absorbed by the water.

The pipe coils are accessible only with difficulty, which makes cleaning and inspecting inconvenient.

The amount of cooling water required is larger than with either of the other two condensers mentioned.

Pipe Surface.—The pipe surface of a submerged condenser may be calculated by the following formula:

$$S = \frac{H \times W}{k \times [t - (t_1 + t_2) \div 2)]}$$

S = required pipe surface in square feet.
H = latent heat in B. T. U. at the temperature t.
W = number of pounds of ammonia passing through the compressor per hour.
t = temperature of ammonia in degrees F. at condenser pressure.
t_1 = temperature of water in degrees F. leaving condenser.
t_2 = temperature of water in degrees F. entering condenser.

k = number of B. T. U. one square foot transmits in one hour for each degree F. difference in temperature between the ammonia inside of the condenser and the average temperature of the cooling water entering and leaving the condenser respectively. For submerged condensers, it may be taken as 30 B. T. U.

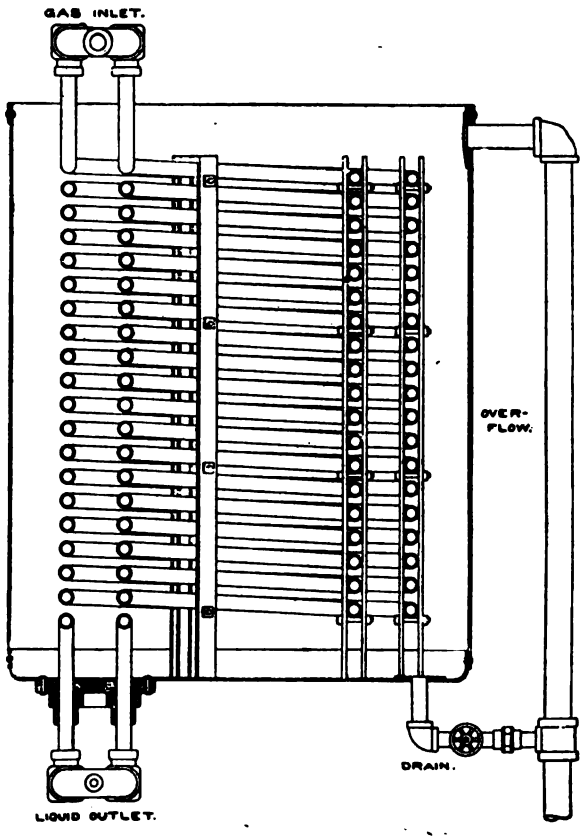


Figure 182. Submerged Condenser.

Example.—A submerged condenser is used in connection with a 50-ton machine working between 25 and 156 pounds gauge pressure, the water entering at 65° F., and leaving at 85°F. How many square feet of pipe surface must the condenser have?

Solution.—First the number of pounds of ammonia which

pass through the expansion valve per hour are calculated. This is done according to formula given on page 506, as follows:

$$W = \frac{288{,}000 \times 50}{[560.9 - (86 - 12) \times 1.1] \times 24} = 1{,}251.$$

1,251 pounds of ammonia are circulated per hour. Consequently, the pipe surface is found according to the formula on page 521 as follows:

$$S = \frac{492.7 \times 1{,}251}{30 \times [86 - (85 + 65) \div 2)]} = \frac{492.7 \times 1.251}{30 \times 11} = 1{,}868.$$

There are 1,868 square feet of pipe surface necessary.

Quantity of Cooling Water.—The quantity of cooling water necessary for a submerged condenser can be calculated in gallons per minute according to the following formula:

$$g = \frac{H \times W}{60 \times (t_1 - t_2) \times 8.33}$$

g = gallons of water per minute.
H = latent heat of ammonia at condenser pressure.
W = number of pounds of ammonia circulated per hour.
t_1 = temperature in degrees F. of water leaving condenser.
t_2 = temperature in degrees F. of water entering condenser.

Example.—How many gallons of cooling water are necessary per minute for a submerged condenser connected to the ice machine considered in the preceding example?

Solution.—According to preceding example, the amount of ammonia circulated per hour is 1,251 pounds. The latent heat H is 492.7, and the temperatures t_1 and t_2 are 85° F. and 65° F., respectively. We therefore have:

$$g = \frac{492.7 \times 1{,}251}{60 \times (85 - 65) \times 8.33} = 61.69.$$

There are 61.69 gallons of cooling water necessary per minute.

In submerged condensers it very seldom happens that the temperature of the water increases 20° F., unless the tank and coils are narrow and high. The author has found a great number of cases, and they may be assumed to represent average conditions, in which the water hardly increased 10° F. unless its temperature on entering was lower than 60° F. If, in the above example, the temperature of the water increased only 15° F., then the gallons of water required per minute would be $\frac{61.69 \times 20}{15}$ = 85.59 gallons.

Hendrick's Condenser.

This condenser belongs to a certain extent to the class of submerged condensers, but differs from the ordinary kind, inasmuch as the water pipes are submerged in the ammonia.

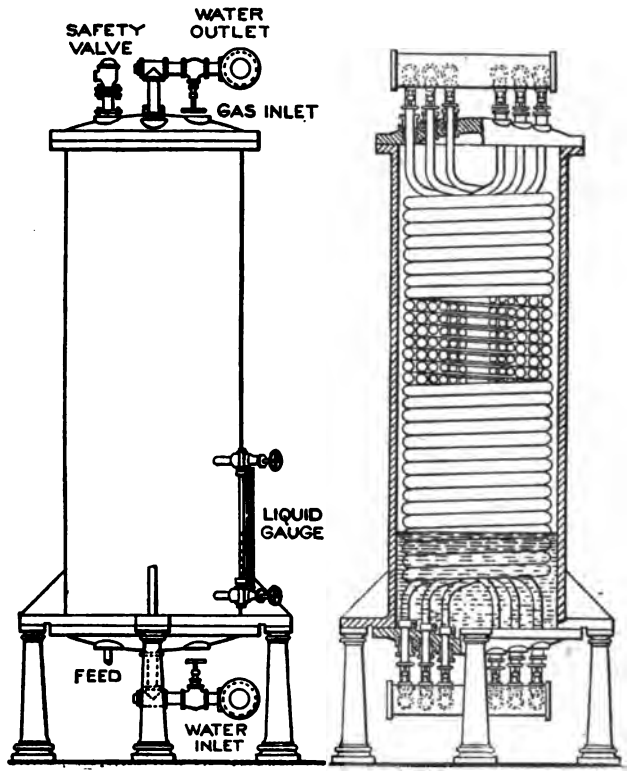


Figure 183. Hendrick's Condenser.

The device, which is illustrated by Figure 183, consists of a strong iron shell standing vertically and containing extra heavy spiral pipe coils, 1¼-inch pipe being usually employed. The ends of the pipes project through the heads of the shell and are connected by headers. The projected ends are kept tight by means of stuffing boxes suitably packed.

The ammonia gas is discharged into the upper part of the shell, which serves at the same time as a liquid receiver, and the liquid is taken out at the bottom. The shell is equipped with a

gauge glass, which indicates to what height the apparatus is filled with ammonia.

The water enters the spiral coil at the bottom, thus rendering cooling more effective. As a result the liquid leaves the device at almost the temperature of the entering water.

It is evident that a condenser of the above construction must be very efficient, since every particle of liquid will immediately drop to the bottom after having been produced, thereby keeping the condensing surface constantly in contact with ammonia vapor.

Atmospheric Condenser.

In the atmospheric condenser, the coils are arranged in vertical sections connected to headers, some having intermediate headers, as shown in the illustration.

Each section is, or at least should be, provided with a stop cock at the inlet and outlet, so that the section may be cut out if necessary.

The water is showered over the vertical pipe sections, cooling being greatly assisted by the surrounding air, the more so, the stronger the current. Therefore, it is always advantageous to place condensers of this type where they can profit from a current of air.

In these condensers, the so-called counter-current cannot be obtained, and the liquid ammonia which leaves the condenser is, therefore, not cooled as low as the temperature of the cooling water would permit, if a counter-current existed. For ammonia machines, this loss is comparatively small, as liquefication is effected considerably below the critical temperature; for carbon dioxide machines, however, the loss proves a serious one, and they are, therefore, seldom used for these machines.

Under average conditions, each section, consisting of 24 2-inch pipes, each 20 feet long, has 10 tons capacity. Manufacturers, however, often claim 12.5 tons and as much as 15 tons per section. This is in most cases too high. Not more than 10 tons should be expected where the plant is operated economically, unless the water is low in temperature, i. e., lower than 65° F.

The condensers should be kept free from accumulation of mud and scale, and if it should prove difficult to remove the latter with ordinary tools, a pneumatic hammer with a chisel of

semi-circular form so as to fit the pipes, should be used. With a tool of this kind, the coils can be freed from scale very rapidly.

By good judgment and a thorough study of prevailing conditions and circumstances, a good idea of the life and working conditions of any part of the condenser may be obtained

The upper header of the condenser should be provided with a purge cock to release air or other permanent gases which may accumulate in the pipes, and which, if they remained in the coils, would reduce the efficient condenser surface by occupying the space intended for the ammonia. The general result would be higher condenser pressure and consequently the expenditure of more power for a given amount of refrigeration to be produced.

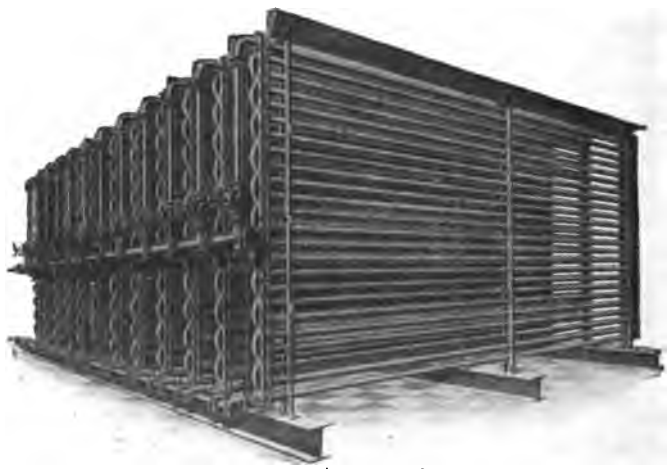


Figure 184. Atmospheric Condenser.

The task of showering water over these coils in the smallest possible quantity, but sufficient to perform the necessary cooling, and of distributing this small amount uniformly over the entire surface is by no means an easy one. The device employed for distributing must be simple and should not become clogged easily. Triangular-shaped, perforated pans are commonly used for the purpose. To prevent spattering of the water in passing from pipe to pipe, to assist distribution and to present a large surface to the air, which aids in cooling, flat perforated strips of metal are sometimes attached to the lower

part of the pipes. The condensers are usually placed in a pan made of cement, asphalt or iron, the outlet of which should be provided with a sufficiently large strainer. Cement bedded on concrete, if properly made, gives best satisfaction.

Pipe Surface.—The formula for the calculation of the proper pipe surface is the same as for submerged condensers, page 521, the only difference being the value of the constant k. While this was 30 B. T. U. for a submerged condenser, it is higher for a condenser of the atmospheric type, viz., 48 B. T. U., because the water is in better circulation.

Example.—How many square feet of pipe surface must an atmospheric condenser have, which is used in connection with a 50-ton ice machine, working between 25 and 156 pounds gauge pressure, the water entering at 65° F. and leaving at 85° F.

Solution.—The number of pounds of ammonia circulated per hour amounts to 1,251 pounds, as will be perceived from the corresponding example of a submerged condenser. The pipe surface therefore is the following:

$$S = \frac{492.7 \times 1{,}251}{48 \times [86 - (85 + 65) \div 2]} = \frac{492.7 \times 1{,}251}{48 \times 11} = 1{,}168.$$

1,168 square feet of pipe surface are required.

Quantity of Cooling Water.—In atmospheric condensers, one gallon of water should be allowed per minute, for every ton of refrigeration produced in 24 hours, if the temperature of the water increases 30° F. Therefore, the water consumption of an atmospheric condenser for a 50-ton machine with an increase of 20° F. in the temperature of the cooling water would be:

$$\frac{50 \times 30}{20} = 75 \text{ gallons per minute.}$$

If the water increased only 15° F., it would require:

$$\frac{50 \times 30}{15} = 100 \text{ gallons per minute.}$$

It is of common occurrence in atmospheric condensers that the temperature of the cooling water increases 20° F. and more, and it is for this reason that these condensers use considerably less water than the submerged condensers.

Block's Atmospheric Condenser.—Mr. Louis Block of New York has recently brought out a condenser, in which the

ammonia gas is allowed to enter at the bottom of an ordinary atmospheric condenser, and the liquid outlet is at the top. At the inlet of each section is placed a valve, which is only a trifle open, i. e., just enough so that the pressure is not higher in the discharge pipe than in the condenser. The condenser is of course full of liquid, and the gas entering at the bottom seems to be condensed at once, in the same manner as steam is rapidly condensed when entering water. The heat transmission is therefore between liquid ammonia and the cooling water, and a square foot is thus more effective than between gaseous ammonia and cooling water. It is claimed that these condensers do nearly twice as much work as the ordinary atmospheric condenser.

Double-Pipe Condensers.

In double-pipe condensers, the water pipe is inside the ammonia pipe. The cooling water enters at the bottom and leaves at the top, while the ammonia travels in the opposite direction, entering at the top and leaving at the bottom; thus a complete counter-current is produced. Figure 185 illustrates a double-pipe condenser.

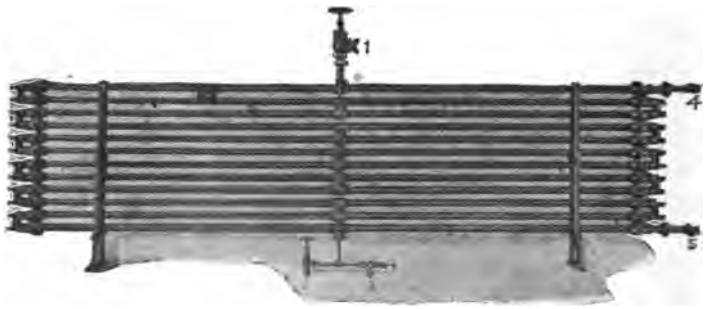


Figure 185. Double-Pipe Condenser.

These condensers are more efficient than the submerged and atmospheric types, provided the water is of such a nature that it does not deposit its ingredients easily or in large quantities in the water pipes.

The life of these condensers is considerably longer than that of the other two types, they can be placed anywhere, they require no pan, the water can be delivered to any desired place after having left the coils without its being pumped again, it

allows of making a reliable test for determining the amount of refrigeration produced by the machine, etc.

The condensers should not be made too long, to avoid sagging of the pipes in the center, which would prevent a free flow of the liquid ammonia. They should have large outlets and a large liquid receiver.

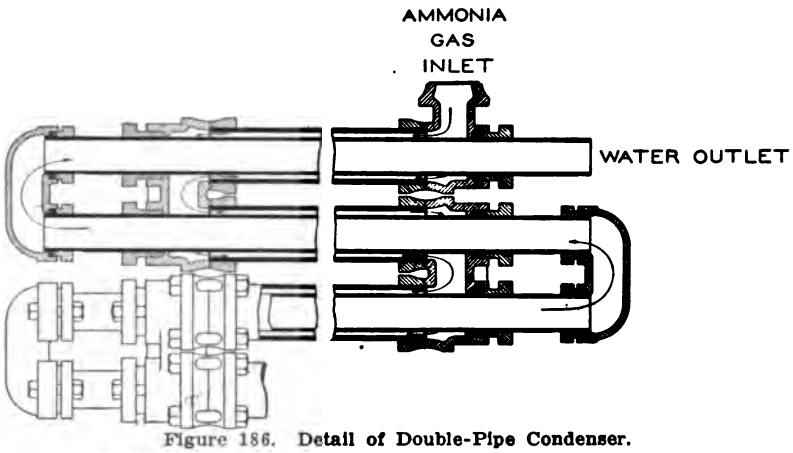


Figure 186. Detail of Double-Pipe Condenser.

Pipe Surface.—A good rule is to allow 10 tons of refrigeration for each section of 12 pipes height, consisting of 2-inch (outside) and 1¼-inch (inside) pipe, the pipes being 20 feet in length.

Water Consumption.—A double-pipe condenser uses a little more water than the atmospheric condenser. The reason for this is that with an atmospheric condenser a certain amount of water is evaporated, while flowing over the pipes and through the air, thus producing a greater cooling effect.

Cooling Towers.

Where the water which must be used for cooling is high in temperature and insufficient in quantity, cooling towers may be employed. If made of proper size, these give very good satisfaction. In cooling towers, the water is used over and over again, the air being in reality the cooling agent and the water only the intermediary. There is a small loss of cooling water, about 10% of the total quantity, which is mainly due to evaporation.

The principle of cooling water by means of air is based upon the fact that moderately dry air absorbs moisture rapidly and causes a corresponding evaporation of water with which it is in contact. As a consequence, the water, by its own evaporation, is lowered in temperature.

The absorption of moisture by air becomes gradually slower until the air is saturated with moisture, at which point absorption ceases altogether. It is evident, therefore, that the better the circulation of air over the water surface, the more rapid the evaporation of the latter, since comparatively dry air always comes into contact with the water.

Figure 187. Hart Cooling Tower.

The capacity of air for carrying moisture increases with the temperature. Air saturated with moisture at 62° F. holds about 0.000881 of a pound of vapor per pound of air, while at 72° F. it holds about 0.001221 of a pound. At 202° F. the amount of moisture in saturated air is approximately 100 times larger than in saturated air of 32° F.

To vaporize a pound of water at atmospheric pressure requires the abstraction of 970.4 B. T. U., and as remarked above, the heat required must be taken from the remaining water, whose temperature is consequently decreased. Since a body with a lower temperature than its surroundings receives heat from the latter,

cooling can only continue as long as the same amount of heat as is absorbed by evaporation is supplied by the surroundings. Evaporation comes to a standstill when, with the lowering of temperature, the elastic force of the vapor formed, and consequently its density, approaches that of the vapor contained in the surrounding air, or condensation and evaporation are at an equilibrium.

There is a maximum density for each temperature and hence a maximum pressure which the vapor exerts. From this, it follows that the rate of evaporation is proportional to the difference between the elastic force of the vapor at the surface of the liquid, and that of the vapor actually present in the surrounding air;

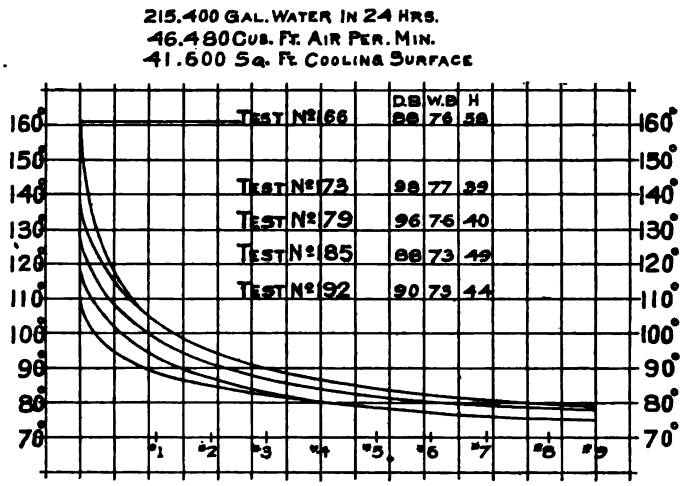


Figure 188. Cooling Tower Test.

evaporation will be the greater, the larger this difference. The efficiency of a cooling tower depends, therefore, on the amount of water surface exposed to the air, the amount of air brought in contact with this surface, and the difference in pressure of the vapors at the water surface, and in the surrounding air.

The degree of cooling depends on the humidity of the air, which means that the drier the air the lower a temperature will be reached in cooling.

When water must be cooled for condensing engines and for ammonia condensers with a cooling tower, it is advisable to use

two *separate cooling towers,* one for the steam condenser and one for the ammonia condenser. Each tower will serve its purpose much better, and each condenser will get water suitable for its purpose. If only one tower is used, the steam condenser receives water which is colder than necessary and the ammonia condenser receives water which is much warmer than it should be.

The degree of cooling should be ascertained by a wet bulb hygrometer, as only in this way can the efficiency of the cooling tower be observed in the proper manner. The hair hygrometer is unreliable for this purpose. Marvin's hygrometer is better, and the recording hygrometer is the best.

Test on Cooling Tower.—A summary of several tests on a cooling tower made by the Ruemelli-Dawley Manufacturing Co. is shown in Figure 188. Referring to the chart, D. B. stands for dry bulb, W. B., wet bulb, and H., humidity. The figures at the bottom, numbers 1 to 9, represent the various relative positions of thermometers during the tests, thermometer No. 1 being near the top.

It is interesting to note that by far the greater portion of the cooling is done between entrance of the water and the point represented by the position of thermometer No. 3, in fact the larger part of the cooling is done before the water reaches thermometer No. 1. For instance, in the test where the initial temperature of the water was 160° F., it was cooled to about 104° F. at the time it reached the position represented by thermometer No. 1, whereas in the rest of the cooling operation, it was only cooled from 104° to 78° F.

MISCELLANEOUS APPARATUS.

Liquid Receiver.

After the ammonia is condensed to a liquid it flows into the so-called liquid receiver, which is generally a strong steel vessel, from 12 to 24 inches in diameter, according to the size of the plant.

It is generally made to hold about 5 pounds of ammonia per ton of refrigeration, and may be either horizontal or vertical, the latter being preferable. It should be provided with a gauge glass to indicate at all times the amount of ammonia in the system.

When all the refrigerating coils are in operation, the liquid receiver should be from ½ to ¾ full.

The receiver also serves as an additional oil trap, inasmuch as the oil, being heavier than the ammonia, will settle on the bottom, where its presence can readily be noted in the gauge glass, and whence it may be withdrawn through an outlet provided for this purpose.

Figure 189. Liquid Receiver.

Storage Drums.

In a direct expansion plant, a receiver of the above-mentioned size is too small in case repairs are to be made on the refrigerating coils, and it is customary to insert a large storage drum between the liquid receiver and the condenser, so that in case of necessity all the ammonia can be stored in the condenser, storage drum and liquid receiver.

The storage drum should have the outlet at the bottom, so that the liquid flows through without being in any way retarded in this drum, otherwise a larger charge of ammonia than actually necessary would be required.

Oil Trap.

The oil trap is similar in construction and is based on the same principle as an oil extractor in a steam line. It is placed between the compressor and the condenser, and the compressed ammonia vapor passes through it, entering at the side and leaving at the top, and depositing any foreign substances which would be carried along into the condenser.

A *check valve* should be inserted in the discharge line near the condenser to hold back the ammonia in case of accident to the compressor.

Most refrigerating machines are provided with cross connections, to enable the engineer to reverse the action of the com-

pressor in such a way as to exhaust the condenser and discharge into the refrigerating coils by opening and closing the proper valves. This combination is called the *by-pass.*

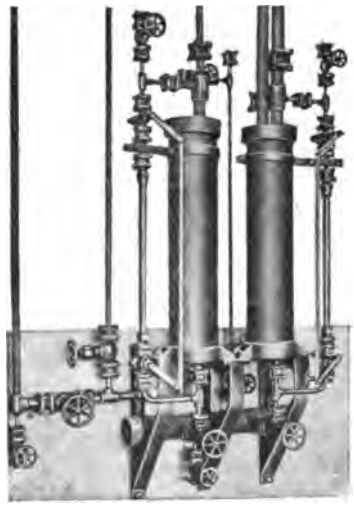


Figure 190. Oil Trap and Liquid Receiver.

Ammonia Valves and Fittings.

Ammonia valves and fittings are made in various styles and kinds. Nearly every ice machine manufacturing company designs and makes its own valves and fittings. It is entirely beyond the scope of this book to describe these, and for information on this subject reference must be made to the catalogues of the respective manufacturers. It may be unnecessary to say that strength, reliability and durability are the main features that govern their value.

STARTING AND OPERATION OF COMPRESSION PLANT.

Before starting a new compression plant, it is first necessary to test it by means of air pressure, the condenser being usually tested at 300 pounds, the refrigerating coils at 200 pounds pressure. In case the slightest leak is observed, after the pressure has been on for 12 to 24 hours, this must be remedied before the system can be charged.

If the system proves tight, air and moisture must be removed as completely as possible. The system is generally provided with the necessary valves for discharging the air, which is removed by starting the compressor with the suction stop valve and the expansion valves open and the discharge stop valve closed.

There is usually a pump-out connection or a cross connection between the discharge stop valve and the compressor, which must be opened to let the air, which the compressor discharges, escape.

When the compressor has removed the air from the coils as thoroughly as possible, the pump-out connection and the expansion valves are closed, while the liquid valve between the liquid receiver and expansion valves and the main discharge stop valve are opened.

The ammonia is then admitted on the suction side, wherever the necessary connection has been made. This varies with the different machines; in the wet compression machines, for instance, the connection is near the compressor, while in dry compression machines it is as far away from the compressor as possible. In the former it makes no difference where the ammonia enters, in the latter, however, it would be harmful should the ammonia enter almost directly into the compressor.

The valve on the drum should always be opened gradually, never suddenly. When the lower part of the drum shows frost it is empty, provided it had been placed in a slightly slanting position, with the outlet at the lower end.

Each drum bears a tag showing the brutto and tara weights of the drum. These weights should be checked by weighing each drum when still full and when emptied.

In order to get out all the ammonia, a vacuum must be drawn on the drum, which can be done the more easily, the nearer the drum connection is to the compressor.

When one drum is empty, another one should be put on in the same manner, and this continued until the system is properly charged.

During all this time water should run over the condenser.

When all expansion valves are opened and properly adjusted, the gauge glass of the liquid receiver should indicate that the latter is ½ to ¾ full.

The suction pressure in the expansion coils, if air surrounds

the pipes, should be kept so as to correspond to a temperature from 15 to 20° F. lower than the required room temperature. Where brine is employed, a difference of from 5 to 10° F. is sufficient, depending upon whether the brine is in motion or not.

The condenser pressure should not exceed that corresponding to a temperature from 5 to 8° F. higher than the temperature of the water leaving the condenser.

The condenser coils should, as much as possible, be kept free from scale and slush, and from time to time should be relieved of air and other foreign gases.

With atmospheric condensers, the water should be evenly distributed over the entire condenser surface, so that no dry spots are noticeable.

The largest quantity of water a gutter can distribute over a condenser of the atmospheric type, without wasting water, is 2⅓ gallons per foot of gutter.

The condenser should be placed where a current of air is naturally circulating, which assists cooling greatly.

The expansion coils should, as far as possible, be kept free from corrosion.

The expansion valves require careful adjustment so that they feed exactly the amount of ammonia evaporated in each coil if a dry compression machine is used, and only a little more in a wet compression system. In the former, the amount of ammonia entering the coils can best be regulated according to the temperature of the compressor piston rod, which should be at blood heat. In other words, by touching the piston rod the engineer can tell whether or not the proper amount of ammonia enters the refrigerating coils.

If the rod becomes warmer, more ammonia should be turned on, and if the rod gets colder, the expansion valves should be checked slightly until the rod is at blood temperature.

In the wet gas compressor, the discharge pipe, instead of the piston rod, serves as the guide for determining the proper amount of ammonia. Operated in this manner, the compressor, if properly designed and in proper condition, will give the highest efficiency; if it fails to do so, the piping system is at fault, i. e., there is either too much or too little refrigerating surface. The former

manifests itself in a low, and the latter in a high suction pressure. The condenser surface may also be insufficient.

All traps on the expansion side should be inspected from time to time and relieved of all substances other than ammonia.

The coils should be examined at least once every two years in order to ascertain if the inside has become coated with oil. Should this be the case, each coil must be blown out separately with heated air.

All stuffing boxes should be kept tight and well packed.

Where the pipe surface in the cooling rooms is insufficient, the air should be set in motion by mechanical means, i. e., by electric fans placed at the same height as the coils. This is also an excellent remedy against wet rooms.

The advantages derived from turning off the coils of the refrigerating rooms from time to time, or stopping the machine for some time, are explained elsewhere.

When stopping the machine, it is not necessary to pump a vacuum on the suction side. It is necessary to shut the liquid inlet valve and stop the machine when the suction gauge shows from 5 to 10 pounds less than the usual pressure. There is no necessity for closing either the suction or the discharge stop valve, unless the machine is to be stopped for a considerable length of time.

If the system must be opened for some reason or other, a vacuum need only be pumped on the suction side.

Before starting the compressor, it should be seen to that all the valves in the discharge line are open and that water is flowing over the condenser. When the suction valve begins to move backwards, the liquid valve should be opened.

In single-acting machines the stuffing box should be tightened after the machine has been stopped, and loosened before the machine is started again. This must be done because the stuffing box is very short and designed to withstand suction pressure only.

When the compressor is in operation, the hands on the gauges should move slightly back and forth. This insures against the gauge being stopped off or the pipe being clogged.

Before opening an ammonia joint, a blow-out valve must be open, in order to be absolutely sure that there is no more pressure.

Ammonia coils should never be blown out in a room where

goods, which are liable to be spoiled, are kept. A hose may easily be attached to the pipe and led outside.

No more oil should be used in the compressor than absolutely necessary.

When removing the slack from the bearings of a double-acting compressor, it becomes necessary to re-adjust the length of the compressor rod so that the piston will not travel closer to one head than to the other.

With horizontal compressors, it becomes necessary to line up the piston from time to time, i. e., to adjust the piston so that it is exactly in the center. This is essential with oval pistons. In some machines, if the piston hangs down only slightly, and if the original clearance was small, the piston is liable to strike the head at the bottom. Many compressors have been wrecked in this manner.

Compressors with oil chamber stuffing boxes should have the oil chamber connected to the suction pipe, and the connections should always be kept open while the compressor is in operation. If this precaution is not observed, it may happen that the rods become hot and gradually heat the oil. If the packing fits tightly, the oil may not have sufficient room for expansion and the stuffing box is liable to burst. Much damage has been done in this manner.

CARBON DIOXIDE MACHINES.

Construction.

The construction of a carbon dioxide plant is similar to that of an ammonia compression plant, the machine consisting in the main of compressor, condenser, and refrigerator. The *compressor,* which need only be $^1/_5$ to $^1/_6$ the size of an ammonia cylinder for the same amount of refrigeration, is made of steel or semi-steel, and in some cases is provided with a jacket through which the return gas passes, this arrangement serving at the same time to cool the compressor and to give additional strength to it. The wearing parts are made of forged steel and are interchangeable.

The packing of the compressor or piston rod consists of leather cups or cast iron rings. Glycerin or a special oil is forced in between stuffing box and rod at a somewhat higher pressure than the suction pressure, this method being somewhat similar to the one practiced on the Linde ammonia machine, with the exception that oil instead of glycerin is used. Any glycerin or oil

passing into the compressor beyond the quantity necessary, goes with the gas through the delivery valves. In order to prevent the glycerin or oil from entering the system, a trap is inserted in the discharge line in which the oil sinks to the bottom, and can be drawn off from time to time.

The *condenser* consists of extra heavy wrought iron coils submerged in a tank filled with water or of the double-pipe type. Double-pipe condensers and coolers are usually employed in connection with carbon dioxide machines with the same success as in connection with ammonia machines.

The construction of carbon dioxide machines is shown in Figures 191 and 192.

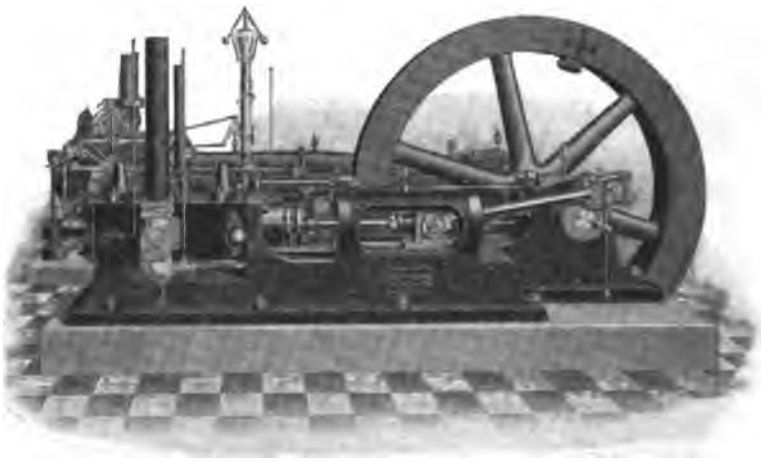


Figure 191. Carbon Dioxide Machine (Kroeschell).

Cycle of Operation.

The cycle in the carbon dioxide machine is practically the same as in an ammonia machine with dry compression.

Advantages of Superheating.

The carbon dioxide, at the temperatures commonly used in practice, is nearly always in a state of saturation on the suction side, whereas, on the compression side it may be partly saturated and superheated at the same time if the vapor enters the compressor wet, or entirely superheated if the vapor enters the compressor in a dry condition. The latter is the case when the

temperature of the condenser water is above the critical temperature of the carbon dioxide.

As the critical temperature of carbon dioxide is very low, viz., 88° F., it becomes necessary in the majority of cases, to work the gas in the superheated state.

In former years, it has been argued quite frequently that the efficiency of the carbon dioxide machine would be greatly reduced if the temperature of the condenser water were above the critical temperature of the gas, and that the actual performance of the machine under such circumstances could not be any greater than that of a machine working with a permanent gas, such as an

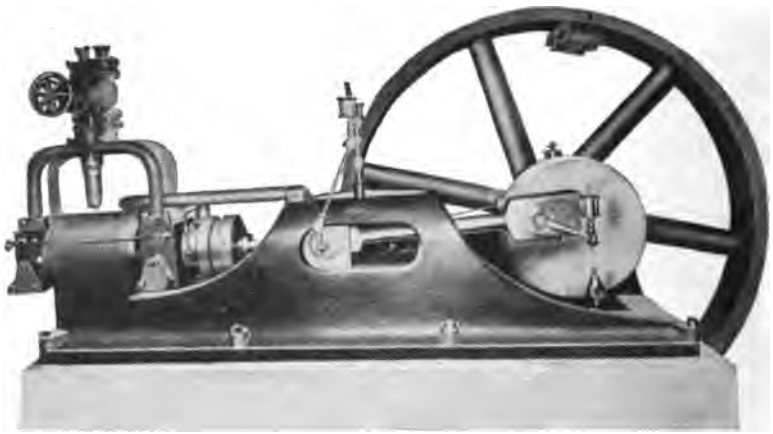


Figure 192. Carbon Dioxide Machine (Am. Carbonic Machry. Co.).

air compression machine; in other words, the refrigerating effect would only be that of a compressed gas expanded against resistance, which would be very small, as only the sensible heat would be given off and absorbed again in the refrigerator. Consequently, a considerably larger compressor would be necessary, as a greater volume of gas would have to be handled and the gas would have to be compressed and expanded several times in order to produce the same refrigerating effect as if the sensible plus the latent heat had been withdrawn and absorbed during one cycle.

This theory would hold good if the critical temperature of carbon dioxide were considerably lower than the average temperature of the cooling water, such as is the case for instance with atmospheric air, whose critical temperature lies at 220° F. below zero. In reality, however, the critical temperature of carbon dioxide, 88° F., and the average temperature of cooling water are only a few degrees apart, and consequently it merely requires an additional final cooling of a few degrees with a substance that is colder than the cooling water. This is generally done with the carbon dioxide gas itself coming from the refrigerator, or by return brine, or by water which has been sufficiently cooled by either brine or carbon dioxide.

The more heat that can be removed from the gas by the cooling water the better, and theoretically it is therefore essential that compression should be performed in the carbon dioxide machine in the superheated state, whereby the temperature is elevated much higher than if compression took place in the saturated condition of the gas. This gives the cooling water a better opportunity to absorb the heat of the condenser, since the difference in temperatures between the water and the gas is greater.

This theory, which makes superheating advisable, is said to hold good in practice. Furthermore, as there are no mechanical difficulties, there is no reason why superheating should not be practiced.

Expansion Cylinder.

Zeuner pointed out that an expansion cylinder should be inserted, to convert into mechanical work, that energy which is stored up on the high pressure side when the pressure is lowered for the refrigerator, which would complete the reversible cycle. However, on account of the small amount of liquid which circulates in a refrigerating machine, this is not practical, and it has been proven in actual practice, the reason being that the loss by friction in such a cylinder almost balances the gain in work.

Advantages of Low Temperature of Carbon Dioxide Before Expansion.

When the refrigerating liquid passes the expansion valve, its temperature necessarily drops, and a certain amount of heat is spent to bring about this drop in temperature. The greater the difference between the temperatures of the liquid before and after

passing the expansion valve, the more heat is lost by this reduction in temperature, and where the specific heat is high and the latent heat of evaporation low, as for instance in carbon dioxide, this involves a serious loss of refrigerating capacity. In the case of ammonia, the loss is less serious, because, although its specific heat is higher than that of carbon dioxide, its latent heat is considerably higher, amounting at atmospheric pressure to 588.5 B. T. U., that of carbon dioxide being only 124 B. T. U. Thus it is at once apparent that if any of this heat is spent for other purposes than for refrigerating, the loss must be considerably more serious with carbon dioxide than with ammonia.

Professor Linde found that the refrigerating effect of a carbon dioxide machine was reduced 50% when the temperature of the carbon dioxide was 92° F. before passing the expansion valve in one case, as compared to another where the temperature was 53° F., the temperature beyond the expansion valve having been the same in both cases. This shows plainly that the liquid should be deprived of as much heat as possible before its pressure, and consequently temperature, is reduced. It would therefore be advantageous if the return gas be made to cool the liquid before it is expanded.

Range of Pressure.

The range of pressure is much higher in a carbon dioxide machine than in any other, and if any leakage should occur, the carbon dioxide escapes very rapidly. The pressures carried in the machines vary from 250 to 400 pounds in the refrigerator, and from 800 to 1,200 pounds in the condenser.

To operate, for instance, between 5° F. and 64° F. requires a suction pressure of 319 pounds and a pressure in the condenser of about 770 pounds.*

In modern designs most of the former mechanical difficulties have been overcome, so that there is no danger of leakage.

Comparison Between Ammonia and Carbon Dioxide Machines.

The manufacturers of ammonia compression machines will always mention the high latent heat of ammonia, those who manufacture sulphur dioxide machines will dwell on the advantages of the low pressure under which their machines operate, while the carbon dioxide machine manufacturers will claim the smallest

* Based on Amagat's table.

size machine for producing a certain amount refrigeration. These claims are only justified with certain limitations, for the one machine does not use less fuel because its refrigerating fluid possesses a larger latent heat, or the other because it works with less pressure, or the third because its compressor is smaller. Each machine has its peculiar advantages and disadvantages, which practically equalize each other at the end of the cycle.

The efficiency of the carbon dioxide machine may be slightly smaller than that of an ammonia compressor when the temperature of the cooling water is high, and only under this condition might it be slightly inferior to the ammonia machine. Its advantages, as far as economy is concerned, are based on the cheapness of liquid carbon dioxide as compared with anhydrous ammonia, and its neutrality towards metals, water, meat, beer and other products subjected to cold storage, etc.

The small size of the compressor, which as has been remarked before need only be $^1/_5$ to $^1/_6$ of the size of an ammonia compressor for the same amount of refrigeration, makes the carbon dioxide machine especially valuable for ships, where the question of space and weight is of paramount importance. Furthermore, where very low temperatures are necessary, the carbon dioxide machine must be used.

In the two following tables are compiled the results of some experiments with ammonia and carbon dioxide machines, made by Lorenz and by Linde and Schroeter in Germany.

Tests on Compressors.

Lorenz.

	Carbon Dioxide Compressor.	Ammonia Compressor.
Compressor gas displacement per ton per minute	1,020 cu. inches	6,920 cu. inches
Compressor work in effective horse-power	50.61 H. P.	62.65 H. P.
Temperature of water entering submerged condenser	53.75° F.	58.6° F.
Temperature of water leaving submerged condenser	70° F.	74.7° F.
Water consumption per ton per minute	1.85 gallons	1.714 gallons
Condenser pressure	994 pounds	144 pounds
Evaporating pressure	373 pounds	28 pounds
Actual cooling effect	49.4 tons	57.36 tons
Horse-power required per ton of cooling effect	1.02 H. P.	1.092 H. P.

Tests on Refrigerating Machines.

Linde & Schroeter.

	Ammonia Machine.				Carbon Dioxide Machine.			
Number of test	1	2	3	4	5	6	7	8
Temperature in Brine tank. Degrees F....	21	20.5	20.5	23.4	24.8	23.4	23.4	20
Temperature in Condenser. Degrees F....	70	70	70	95	69	70	72	86
Temperature before expansion valve. Degrees F....	20.3	53	65	83	18	50	62	84
Refrigeration per hour, per horse-power of steam engine. B. T. U.	15,531	14,489	13,979	8,914	15,270	12.664	11,425	5,886

Properties of Saturated Carbon Dioxide Vapor.*

Temperature Degrees F.	Absolute Temperature Degrees F.	Absolute Pressure Pounds per Square Inch.	Gauge Pressure Pounds per Square Inch.	Volume of Vapor in Cu. Ft. per Lb.	Weight of Vapor in Lbs. per Cu. Ft.	Heat of Liquid Above 32° F. B. T. U. per Lb.	Latent Heat of Vaporization B. T. U. per Lb.	Total Heat Above 32° F. B. T. U. per Lb.
— 22	438	213.0	198.3	0.4323	2.32	—25.72	126.13	100.41
— 13	447	248.5	233.8	0.3674	2.73	—21.87	122.67	100.80
— 4	456	288.3	273.6	0.3132	3.19	—17.87	118.86	100.99
+ 5	465	333.7	319.0	0.2674	3.74	—13.73	114.71	100.98
14	474	384.8	370.1	0.2286	4.37	— 9.38	110.12	100.74
23	483	440.2	425.5	0.1952	5.12	— 4.82	105.04	100.22
32	492	502.7	488.0	0.1669	6.00	0.	99.34	99.34
41	501	573.3	558.6	0.1422	7.03	+ 5.17	92.91	98.08
50	510	648.9	634.2	0.1205	8.30	10.76	85.54	96.30
59	519	732.7	718.0	0.1010	9.90	17.01	76.84	93.85
68	528	825.0	810.3	0.0840	11.92	24.21	66.15	90.36
77	537	928.7	914.0	0.0672	14.85	33.19	51.91	85.10
86	546	1,038.0	1,023.3	0.0474	21.09	47.50	26.88	74.38
87.8	547.8	1,060.7	1,046.0	0.0412	24.27	53.77	15.04	68.81
88.4	548.4	1,069.3	1,054.6	0.0346	28.95	61.45	0.	61.45

* Amagat.

Some more comparative figures are given under "Sulphur Dioxide Machines," and from these as well as from the above data, it appears that there is comparatively little difference in the efficiency of the ammonia, the carbon dioxide and the sulphur dioxide machines.

Carbon Dioxide as a By-Product Obtained from the Fermenting Vats in Breweries.

In many breweries, carbon dioxide collecting systems are installed, whereby the carbon dioxide, as fast as it is formed, is collected with a low pressure pump. This pump delivers the gas into purifying vessels, whence it passes into storage drums. From these drums, the gas is either used for carbonating purposes directly, or it is taken into a carbon dioxide refrigerating machine and raised to a much higher temperature and pressure and liquefied. The liquid is then drawn off into strong drums of a commercial capacity of 20 and 50 pounds. These drums are of five and seven inches internal diameter, and 50 inches long. They are made of steel ¼ inch thick. The 20-pound drums usually weigh 70 pounds when empty, and are made of steel which has a tensile strength of 60,000 pounds per square inch. The 50-pound drums usually weigh 100 pounds when empty, and are made of steel, which has a tensile strength of 95,000 pounds per square inch. They are filled to 55% of their water-holding capacity, which allows sufficient expansion for temperatures up to 130° F., to which temperature they might be subjected when lying in the sun and under which condition the pressure rises to about 2,000 pounds per square inch. The drums will usually rupture at a pressure of from 4,500 to 5,500 pounds per square inch.

For calculating storage drums to hold carbon dioxide in the gaseous form at various pressures and temperatures, the following formula may be applied:

$$V_1 = \frac{V \times P \times T_1}{P_1 \times T}$$

V_1 = final volume of one pound of gas at T_1 and P_1.
V = volume of one pound at T and P.
T_1 = absolute temperature in degrees F. of gas at V_1.
T = absolute temperature in degrees F. of gas at V.
P_1 = absolute pressure in pounds per square inch at V_1.
P = absolute pressure in pounds per square inch at V.

Example.—A pound of carbon dioxide gas has a volume of 8 cubic feet at 32° F. and atmospheric pressure. What is its volume at 40° F. and 115 pounds absolute pressure?

Solution— $V_1 = \frac{8 \times 14.7 \times (40 + 460)}{115 \times (32 + 460)} = 1.04$ cubic feet per pound.

Example.—How many pounds of carbon dioxide gas will a tank 4 feet in diameter and 8 feet deep, hold at 60 pounds gauge pressure (75 pounds absolute), and 50° F.?

Solution— $V_1 = \frac{8 \times 14.7 \times (50 + 460)}{(60 + 15) \times (32 + 460)} = 1.625$ cubic feet per pound.

Hence there are:

$$\frac{4 \times 4 \times 0.7854 \times 8}{1.625} = 61.86 \text{ pounds in the tank.}$$

SULPHUR DIOXIDE MACHINES.

The sulphur dioxide machine was one of the first machines that worked successfully with condensable vapor, although sulphur dioxide is not well suited for refrigerating purposes.

Properties of Sulphur Dioxide.

Sulphur dioxide (SO_2) is a colorless gas of a characteristic pungent odor. It liquefies under atmospheric pressure at 14° F. Its critical temperature is 247° F., its critical pressure 1,142 pounds. Its vapor density is 32. The specific heat of the liquid is 0.41, its specific gravity is 1.49 (water = 1), while the specific gravity of the gas is 2.23 (air=1). The specific heat of the gas at constant pressure is 0.1544, and the ratio of specific heat at constant pressure and constant volume is $\frac{0.1544}{0.1229}$ or 1.256. The latent heat of sulphur dioxide is comparatively small; it varies with the temperature, and at atmospheric pressure is 168.5 B. T. U. per pound. A table of the various properties of saturated sulphur dioxide vapor is given on page 547. The values are somewhat different from those of Ledoux' table.

The liquid anhydrous sulphur dioxide has no injurious effect on iron, steel or copper, and any of these metals may therefore be used. If water, however, enters the system, sulphurous acid

is formed, and this destroys iron or brass in a short time. It is therefore necessary to guard against such possibilities.

Properties of Saturated Sulphur Dioxide Vapor.*

Temperature Degrees F.	Absolute Temperature Degrees F.	Gauge Pressure Pounds per Square Inch.	Absolute Pressure Pounds per Square Inch.	Volume of Vapor in Cu. Ft. per Lb.	Weight of Vapor Lbs. per Cu. Ft.	Heat of Liquid Above 32° F. B. T. U. per Lb.	Latent Heat of Vaporization B. T. U. per Lb.	Total Heat Above 32° F. B. T. U. per Lb.
— 40	420	....	3.12	22.715	0.044	—21.33	178.56	157.23
— 31	429	....	4.26	17.121	0.058	—18.83	177.36	158.53
— 22	438	....	5.54	13.177	0.076	—16.29	175.99	159.70
— 13	447	....	7.24	10.307	0.096	—13.72	174.42	160.70
— 4	456	....	9.23	8.223	0.122	—11.07	172.66	161.59
+ 5	465	....	11.79	6.668	0.150	— 8.39	170.68	162.29
14	474	0.07	14.77	5.290	0.189	— 5.65	168.48	162.83
23	483	3.62	18.32	4.328	0.231	— 2.84	166.08	163.24
32	492	7.74	22.44	3.574	0.280	0.	163.48	163.48
41	501	12.70	27.40	2.950	0.340	+ 2.90	160.65	163.55
50	510	18.53	33.23	2.437	0.411	5.85	157.61	163.46
59	519	15.20	39.90	2.036	0.493	8.86	155.36	163.22
68	528	32.87	47.57	1.715	0.585	11.92	150.93	162.85
77	537	41.53	56.23	1.443	0.696	15.03	147.28	162.31
86	546	51.61	66.31	1.218	0.824	18.20	143.80	161.60
95	555	62.83	77.53	1.042	0.964	21.42	139.32	160.74
104	564	85.47	90.17	0.882	1.141	24.68	135.05	159.73

Pressures Required.

The pressure carried in the refrigerator must be very low; at 14° F. it only reaches the atmospheric pressure, and for the usual temperatures required for refrigeration, the pressure must be carried at or below that of the atmosphere. It is therefore evident that if the stuffing box or any other joint shows the slightest leak, air is drawn into the system, destroying the capacity and economy of the machine. If, however, higher temperatures and consequently higher pressures can be kept, these machines work successfully, which fact has been demonstrated many times by practical tests, and is proven by the large number of machines which are in successful use in Germany.

* Cailletet and Mathias.

Size of Compressor.

The volume of the compressor must be nearly three times as great as for an ammonia machine of the same refrigerating capacity.

It has been found in practice that for ice making, there must be drawn into the compressor about 150 cubic inches of vapor in order to abstract one B. T. U.

Efficiency.

In a series of tests made in Germany by Professor Schroeter with a condenser temperature of 78° F., and a suction temperature of 6° F., an ice melting capacity of 21 pounds was obtained per pound of coal, which corresponds to the abstraction of 1 B. T. U. from the refrigerating room for every 3¼ B. T. U. furnished by the boiler.

If a boiler and furnace efficiency of 65% is assumed, then the efficiency of the sulphur dioxide machine referred to the B. T. U. in the coal, would be $\frac{3.25}{0.65} = 5$, which means that for every 5 B. T. U. in the coal, 1 B. T. U. is abstracted from the refrigerating room.

Comparison Between Ammonia, Carbon Dioxide and Sulphur Dioxide Machines.

By comparing the efficiencies of the ammonia, carbon dioxide and sulphur dioxide machines under average conditions, we find that for every B. T. U. abstracted from the refrigerating room the following number of B. T. U. must be furnished by the boiler:

Ammonia machine	3.3	B. T. U.
Carbon dioxide machine.....	3.37	B. T. U.
Sulphur dioxide machine.....	3.35	B. T. U.

It has been assumed in calculating the above that a simple non-condensing Corliss engine is employed which uses about 30 pounds of steam per horse-power per hour.

The above figures are for machines of 50 tons capacity. For other sizes the figures vary according to the difference in power required to drive the compressor. With smaller machines the losses in general are greater.

When calculating and comparing the efficiencies of the three

compressors, i. e., comparing the amount of work done in compressing the fluid to the work expended in the engine, we find:

Efficiency of Ammonia Compressor, 87 to 93%.

Efficiency of Carbon Dioxide Compressor, 88 to 93%.

Efficiency of Sulphur Dioxide Compressor, 89 to 91%.

This shows that there is little difference in the efficiencies of the three machines.

METHYL ETHER AND METHYL CHLORIDE MACHINES.

Methyl ether and methyl chloride are two substances frequently used in France for refrigerating purposes, but not in the United States.

Methyl ether is produced by heating wood alcohol with sulphuric acid. At atmospheric pressure it liquefies at —10° F.

Methyl chloride can be manufactured in various manners. At atmospheric pressure it liquefies at about —10° F.

These two substances have nearly the same pressures at temperatures from —4° F. to +68° F., the pressures being higher at low temperatures than is the case with sulphur dioxide.

The latent heat of methyl ether at 32° F. is 360 B. T. U., its condenser pressure at 68° F. is only 70 pounds, and at 104° F. it is 120.5 pounds, while the suction pressure at —4° F. amounts to 17 pounds. It also has the advantage of being absolutely non-injurious to metals. For methyl chloride, the conditions are similar, and provided the temperatures required are not too low, both substances can be successfully employed for refrigerating purposes. On the other hand they have a number of disadvantages which will always prevent these machines from gaining a large field or from successfully competing with ammonia refrigerating machines.

Among these disadvantages may be mentioned: (1) both substances are highly inflammable and explosive; (2) owing to their low vapor tension, a very large volume must be circulated to obtain a given refrigerating effect, thus abnormally increasing the dimensions of the machine; (3) the cooling water must, for good economy, be very low in temperature, or else very large quantities are required.

PICTET LIQUID MACHINE.

His endeavor to improve refrigerating machines caused Prof. Pictet to search for a refrigerating fluid that would possess a high evaporating pressure and a low condensing pressure. As a mixture of 97% by volume of sulphur dioxide and 3% carbon dioxide seemed to fill the requirements, he designed a machine suitable for this liquid. This machine and its auxiliary apparatus differed in many respects from his former sulphur dioxide machine. In later practice, however, it was found that the mixture of the two substances possessed no material advantages, and the Pictet machines of today are not working with the Pictet liquid but with sulphur dioxide.

With the Pictet liquid it seems that when the machine is in operation, the carbon dioxide has the tendency to remain in the condenser, and that finally the sulphur dioxide alone is the circulating fluid.

INDICATOR DIAGRAMS OF COMPRESSOR.

As has been remarked previously, the performance of the compressor of a refrigerating machine is the reverse of that of the cylinder of a steam engine. In the latter, heat is transformed into work, inasmuch as the steam enters the cylinder at a high temperature, and after having performed its work behind the piston leaves it at a lower one. In the compressor, the gas enters at a low temperature, and the piston in performing the work of compressing the gas, raises its temperature. Since the process is similar, it is also possible to test the performance of a compressor in a similar manner as that of a steam cylinder, viz., by means of an indicator. It is of the utmost importance that indicator diagrams of the compressor as well as of the engine driving it, be taken regularly. The performance of the entire system and the economy of the compressor may thus be ascertained and practically all defects of the system revealed.

The instrument used for indicating a compressor is similar to that employed for a steam cylinder, except that it should be made of steel, because this metal is not attacked by ammonia. The steam engine indicator may, however, be used on the ammonia compressor, provided the indicator is thoroughly cleaned after each test.

Diagrams of the compressor are similar to those of a steam cylinder, except that in case of a compression diagram the lowest point of the curve is that at which the ammonia enters the compressor. From this point the curve rises to the pressure at which the gas is discharged. The curve itself is also somewhat different from the steam compression curve. Figure 193 gives a sketch of an ammonia compressor indicator diagram with the important points and lines designated as follows:

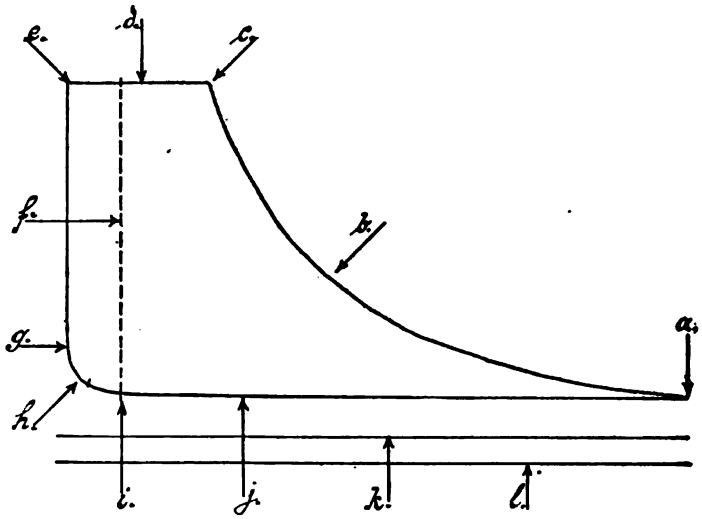


Figure 193. Compressor Diagram.

a = closing of suction valve and beginning of compression.
b = compression curve.
c = opening of discharge valve.
d = discharge pressure line.
e = closing of discharge valve.
f = clearance line.
g = beginning of return stroke.
h = expansion curve.
i = opening of suction valve.
j = suction pressure line.
k = atmospheric line.
l = vacuum line.

Ideal Curve.

The ideal curve of an ammonia or any other gas compressor is the isothermal curve. If this curve could be obtained, the least amount of work would be required for the compression of a gas. It would require, however, a perfect and frictionless compressor, and facilities for removing the heat of compression immediately. Since this is not possible in practice, the real compression curve will always be found to be above the isothermal curve, and should the real curve ever be found to be at or below the isothermal it is a sure sign that either the piston or the suction valve or both leak. The maximum work that would be required for the compression of a gas, in a perfect frictionless compressor would be indicated by the adiabatic curve, which would require that none of the compression heat be removed. Since this is not

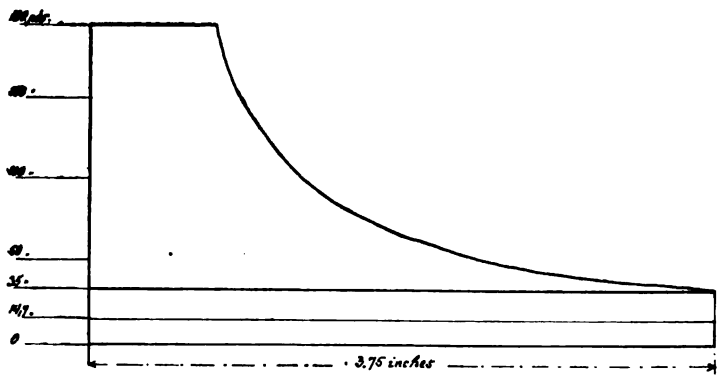


Figure 194. Ideal Compressor Curve.

possible in practice, the real curve for ammonia compression will always be found at or slightly above the adiabatic curve in the early part, and below in the latter part, of the stroke. This is due to the fact that the ammonia gas entering the compressor is colder than the cylinder walls, consequently heat is absorbed by the gas from the walls, but as the compression of the gas increases, its temperature rises and finally gets above the temperature of the cylinder walls, whence heat will be given off by the gas. It does not seem to make any particular difference whether the compressor works wet or dry, which plainly indicates that the idea of ad-

mitting liquid into the compressor, for the prevention of superheating during compression is wrong, as practically no evaporation of the liquid takes place during the greater part of the stroke, such as was theoretically expected. In the final part of the stroke, however, some of the liquid, on account of the existing high temperature and the agitation to which the liquid drops are subjected, does evaporate, so that the gas which was superheated during compression enters the discharge pipe in a more or less saturated state. This fact was mainly responsible for the belief of so many refrigerating engineers in the theory that less work is required for a given quantity of gas if no superheating takes place during compression, hence more refrigeration could be produced for less

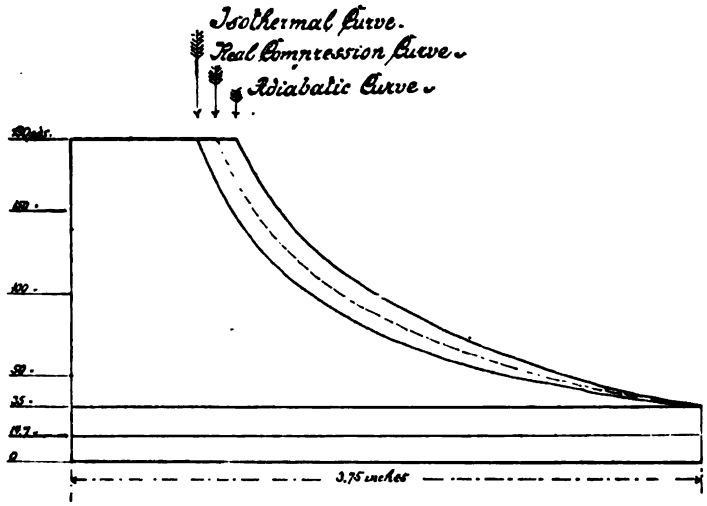


Figure 195. Adiabatic and Isothermal Compression Curves.

energy expended. However, after a more close examination of various compressor diagrams from different machines, the above stated fact was finally brought to light, showing plainly that nothing is gained by wet compression. There is a loss connected with wet compression, for the simple reason that wherever liquid remains in the clearance space it is entrapped between hot walls, and highly agitated, and will therefore rapidly evaporate during the return stroke, whereby the capacity as well as the economy is decreased.

Figure 194 shows an ideal diagram, and Figure 195 a diagram

with the adiabatic curve, the isothermal curve, and an actual compression curve between the two extreme curves.

Construction of Isothermal and Adiabatic Curves.

By the laying out of the isothermal and adiabatic curves on a compressor card, a number of faults may be readily detected, whereas without these curves the diagram is liable to be misleading.

After constructing the two curves on a card, in many cases where the diagram is apparently faultless, it is found that the compressor curve is not where it should be. Leaks through the valves, pistons, or both, are shown in this manner.

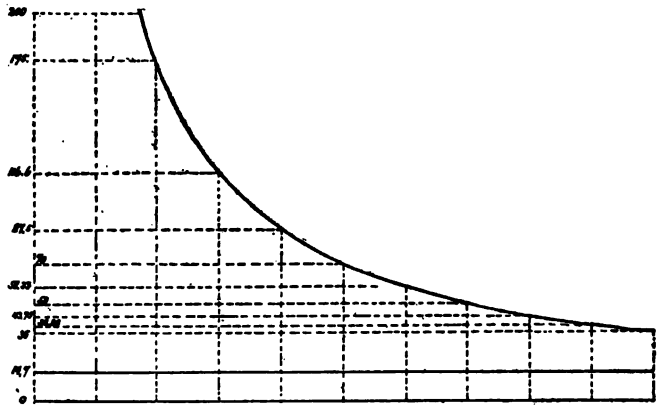


Figure 196. Construction of Isothermal Curve.

Construction of Isothermal Curve.—The isothermal curve or curve of equal temperatures is a common or rectangular hyperbola, which may be constructed according to the following laws:

1. The pressure varies inversely as the volume when the temperature is constant.

2. The product of the volume and pressure is proportional to the absolute temperature.

3. The volume varies as the absolute temperature when the pressure is constant.

4. The pressure varies directly as the absolute temperature when the volume is constant.

If p and v stand for the initial pressure and volume, and p_1 and v_1 for the final pressure and volume respectively, then

$$p_1 \times v_1 = p \times v$$

$$p_1 = \frac{p \times v}{v_1}$$

$$v_1 = \frac{p \times v}{p_1}$$

Figure 196 shows the isothermal curve drawn in accordance with the above formula.

To construct the isothermal curve on an indicator card taken from a compressor, proceed as follows: Draw the vacuum line 14.7 pounds below the atmospheric line, divide the card into 10 or more equal parts, and the corresponding pressure for each volume can be found by applying the equation $p_1 = p \times \frac{v}{v_1}$ Supposing $v = 10$ parts and $p = 35$ pounds absolute pressure, then the pressure p_1 for each part of v which would be v_1, would be:

$$35 \times \frac{10}{9} = 38.88 \text{ lbs.}$$

$$35 \times \frac{10}{8} = 43.75 \text{ lbs.}$$

$$35 \times \frac{10}{7} = 50.00 \text{ lbs.}$$

$$35 \times \frac{10}{6} = 58.33 \text{ lbs., etc.}$$

It can readily be seen from the formula that $p \times v$ is constant, and in the above case is equal to 350, and by dividing this constant by the 10 different volumes the pressure for each volume is obtained. The points corresponding to these pressures are connected with a curved line, this being the isothermal curve.

For the geometrical construction of the isothermal curve, see page 559.

Construction of Adiabatic Curve.—The calculation and construction of the adiabatic curve are more difficult and complicated than of the isothermal curve, because the temperature does not remain constant during compression. The relations between pressure, volume and temperature may be expressed by equations, by the use of which, various points of the adiabatic curve can be calculated, as will be demonstrated below by an example:

The equations are the following:

1. Relation of pressure to volume

$$\frac{p_1}{p} = \left(\frac{v}{v_1}\right)^k$$

2. Relation of volume to pressure:

$$\frac{v}{v_1} = \left(\frac{p_1}{p}\right)^{\frac{1}{k}}$$

3. Relation of pressure to temperature:

$$\frac{p_1}{p} = \left(\frac{T_1}{T}\right)^{\frac{k}{k-1}}$$

4. Relation of temperature to pressure:

$$\frac{T_1}{T} = \left(\frac{p_1}{p}\right)^{\frac{k-1}{k}}$$

5. Relation of volume to temperature:

$$\frac{v}{v_1} = \left(\frac{T_1}{T}\right)^{\frac{1}{k-1}}$$

6. Relation of temperature to volume:

$$\frac{T_1}{T} = \left(\frac{v}{v_1}\right)^{k-1}$$

In the above equations the letters used designate as follows:

p = initial absolute pressure in pounds per square inch.
p_1 = final (or intermediate) absolute pressure in pounds per square inch.
v = initial volume.
v_1 = final (or intermediate) volume.
T = initial absolute temperature in degrees F.
T_1 = final (or intermediate) absolute temperature in degrees F.
k = a factor which varies with the nature of the gas.

The factor k expresses the ratio of the specific heat at constant pressure (c_p) to the specific heat at constant volume (c_v) and can therefore be expressed by the formula $k = \frac{c_p}{c_v}$ Its value for the most important gases is given herewith:

Air .....k = 1.41
Sulphur Dioxide .....k = 1.256
Carbon Dioxide .....k = 1.262
Ammonia .....k = 1.32*
Gaseous Steam .....k = 1.283

For *ammonia,* the relation between pressure and volume is:

$$\frac{p_1}{p} = \left(\frac{v}{v_1}\right)^{1.32}$$

This signifies that the pressures are inversely proportional to the volumes raised to the 1.32 power.

The relation of volume to pressure is:

$$v_1 = \frac{v}{\left(\frac{p_1}{p}\right)^{0.76}}$$

The relation of pressure to temperature is:

$$p_1 = p\left(\frac{T_1}{T}\right)^{4.12}$$

The relation of temperature to pressure is:

$$T_1 = T\left(\frac{p_1}{p}\right)^{0.24}$$

The relation of volume to temperature is:

$$v_1 = \frac{v}{\left(\frac{T_1}{T}\right)^{3.12}}$$

The relation of temperature to volume is:

$$T_1 = T\left(\frac{v}{v_1}\right)^{0.32}$$

The final pressure p_1 or the final volume v_1 or any intermediate pressure or volume is mostly required.

The following example illustrates how the adiabatic curve is laid out (see Figure 197). The initial absolute pressure is 35 pounds, the initial volume 10, i. e., the length of the diagram is divided in 10 equal parts, the final pressure is 200 pounds absolute, hence:

* Frequently given as 1.298.

$p = 35$
$p_1 = 200$
$v = 10$
$v_1 = 9, 8, 7, 6, 5, 4, 3, 2$ and 1.

By transposing the equation on page 557, the following formula for finding p_1 at any v_1 is obtained:

$$p_1 = p \left(\frac{v}{v_1} \right)^{1.32}$$

This formula signifies that the ratio of the volumes is to be raised to the 1.32 power and then multiplied by the initial pressure, which gives the final (or intermediate) pressure.

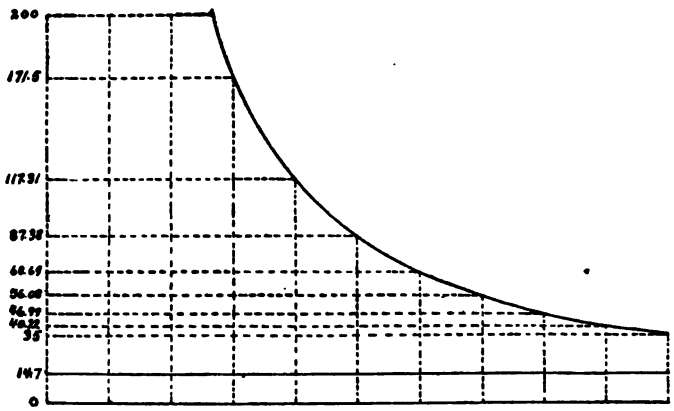


Figure 197. Construction of Adiabatic Curve.

To raise a number to a fractional power requires the use of logarithms, hence the solution for the above example will be as follows:

$$p_1 = p \times \left(\frac{v}{v_1} \right)^{1.32} = 35 \times \left(\frac{10}{9} \right)^{1.32}$$

Log. of $\frac{10}{9} = 0.045757$; $0.045757 \times 1.32 = 0.060399$. The number corresponding to the logarithm 0.060399 is 1.149; $1.149 \times 35 = 40.22$, which is the pressure if 10 volumes are reduced to 9. The following pressures obtained in this manner correspond to their respective volumes:

$\frac{10}{8}$ = 46.99 pounds, $\frac{10}{7}$ = 56.08 pounds, $\frac{10}{6}$ = 68.69 pounds, $\frac{10}{5}$ = 87.38 pounds, $\frac{10}{4}$ = 117.31 pounds, and $\frac{10}{3}$ = 171.5 pounds.

For $\frac{10}{2}$ volumes we would obtain a pressure of 292.99 pounds, but since the final pressure is only 200 pounds, the final volume is not 2, but according to the equation given on page 557 will be:

$$v_1 = \frac{v}{\left(\frac{p_1}{p}\right)^{0.76}} = \frac{10}{\left(\frac{200}{35}\right)^{0.76}}$$

$200 \div 35 = 5.7143$. Log. of 5.7143 is 0.75696, and $0.75696 \times 0.76 = 0.575290$. The number corresponding to the logarithm 0.575290 is 3.7689, and $10 \div 3.7689 = 2.653$, which is the final volume at 200 pounds absolute pressure.

Figure 197 shows the curve drawn in accordance with the above example.

To save time and avoid reference to the table of logarithms, the following values are useful. To find the pressure corresponding to each volume, it is merely necessary to multiply the respective values by the absolute suction pressure as determined on the indicator card.

$\left(\frac{10}{9}\right)^{1.32} = 1.14921$ $\quad$ $\left(\frac{10}{5}\right)^{1.32} = 2.49666$

$\left(\frac{10}{8}\right)^{1.32} = 1.34252$ $\quad$ $\left(\frac{10}{4}\right)^{1.32} = 3.35190$

$\left(\frac{10}{7}\right)^{1.32} = 1.60129$ $\quad$ $\left(\frac{10}{3}\right)^{1.32} = 4.90005$

$\left(\frac{10}{6}\right)^{1.32} = 1.96264$ $\quad$ $\left(\frac{10}{2}\right)^{1.32} = 8.36835$

Geometrical Construction of Adiabatic and Isothermal Curves.*—For the engineer to whom the use of drawing tools is more handy than the use of logarithms or other cumbersome calculations, the following geometrical construction of

* Otto Luhr, American Brewers' Review, Vol. XXII., Page 434.

the adiabatic and isothermal curves may be used (see Figure 198).

To construct the adiabatic curve, first draw the vacuum line ab at a distance of 14.7 pounds below the atmospheric line and parallel to it. Then draw the vertical line ac at the pressure end of the diagram. At the intersection a of these two lines, construct two angles, one 20 degrees from the vertical line, and one 15 degrees from the horizontal line.

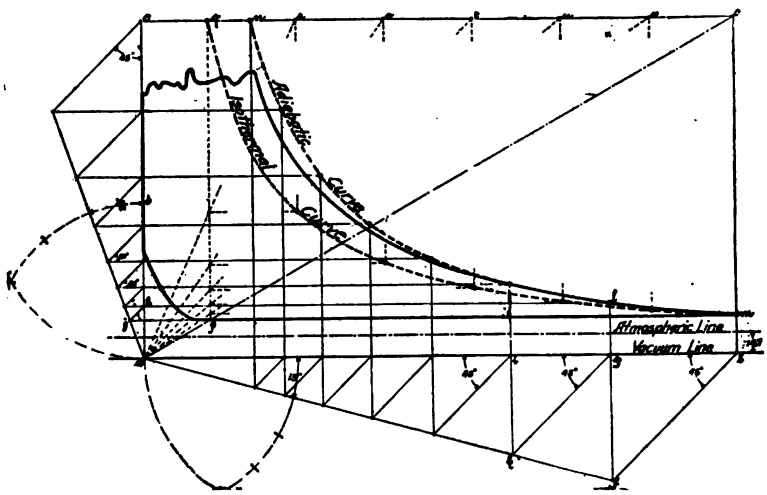


Figure 198. Geometrical Construction of Isothermal and Adiabatic Curves.

The 20 degree angle may easily be constructed as follows: Around point a, draw an arc with the radius ad, which may be of any convenient size, for instance, 3 or 4 inches. With point d as a center and da as a radius, draw another arc through a, and intersecting the arc de at point e. Divide arc de into three equal parts and connect the dividing point nearest to d with the point a, this giving the desired angle of 20 degrees.

The 15 degree angle is located in the same way, excepting that the arc is divided into four instead of three equal parts. Draw a vertical line from the low-pressure end of the diagram to the vacuum line, thus locating point b. From point b, draw a 45° degree angle, extending it until it intersects the 15 degree angle line at f. Then draw a vertical line upward to the vacuum

line, which locates point g. Now draw another 45 degree line downwards to point h and another vertical line to i, and repeat until all the required vertical lines have been constructed. Now extend the base line of the diagram until it crosses the 20 degree angle line at point j, and from this point draw a 45 degree angle line to ac. At the intersection k, draw a horizontal line until it meets the vertical extended line fg at point l. This point l is one of the points on the adiabatic curve. Repeating this method, as shown in the diagram, will locate all the other required points, and by connecting them, establishes the adiabatic curve mn.

The *isothermal curve* is found in the following manner: Draw a line bo perpendicular to the vacuum line and intersecting the horizontal line co at o. The line co passes through the highest point n of the adiabatic curve. Now connect points o and a, the connecting line passing through the base of the diagram at point p. At this point, erect a perpendicular line which meets the line co at point q, which is a point of the isothermal curve. On the line qo, mark off any number of points r, s, t, u and v, which are selected at random. The more of these points that are selected near the high pressure end of the diagram, the more accurately can the curve be constructed. Connect points r, s, t, u and v with point a, and where these connecting lines pass through line pq, draw horizontal lines until they meet the vertical lines passing through r, s, t, u and v. The intersections of these horizontal and vertical lines are points on the isothermal curve, and when connected establish that curve.

Criticism of Indicator Cards.

The following diagrams may serve in comparing faults of the compressor.

Figure 199 shows a card that may be expected if the compressor is in very good condition with re-expansion 5%.

Figure 200 shows a card with *excessive re-expansion* amounting to 15%. This may be due to too much gas of high pressure, or high pressure gas and liquid remaining in the compressor after the stroke is completed. The result will be a re-expansion of the high pressure gas, and if liquid is present and evaporation takes place when the piston begins its return stroke,

the expanding high pressure gas and the evaporating liquid will occupy the space created by the returning piston until the pressure is reduced to that of the suction pressure, before any new gas can enter the compressor. The point where the expansion line reaches the suction pressure line extended, indicates the amount of space the remaining gas occupied, and is thus called re-expansion clearance. It is needless to say that the capacity of the compressor is reduced an amount equal to the re-expansion. Excessive re-expansion can also be produced by leaky discharge valves.

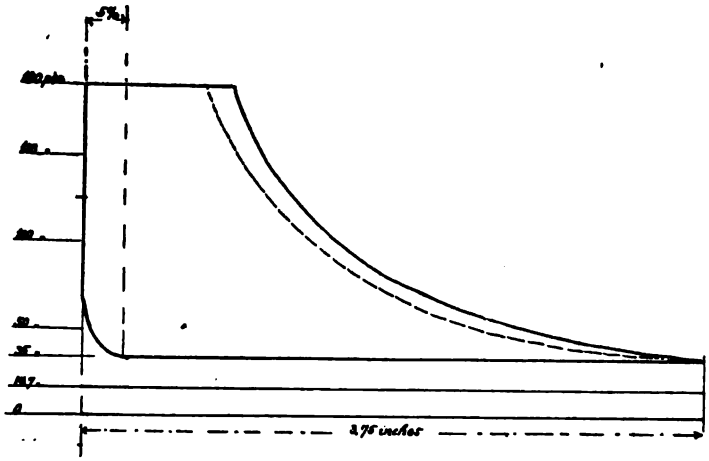


Figure 199. Card of a Good Compressor.

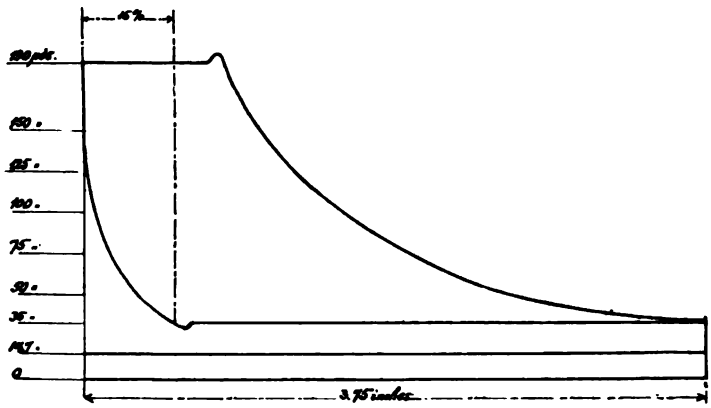


Figure 200. Excessive Re-expansion.

Figure 201 shows the so-called *suction depression,* which may be caused by excessive friction of the gas passing through the expansion coils, or the suction pipe, or the suction valves may be too small. Moreover, if the pressure at the beginning of the stroke corresponds with the proper back pressure and still the compression curve does not rise immediately, then the suction valve sticks or binds, which can be seen plainly in Figure 202. The loss is the same as re-expansion loss.

Figure 203 shows *excessive final pressure* prevailing in the compressor while the discharge valve is open, which may be produced by the gas passage being too small or by the excessive tension of the discharge valves, by which a frequent delay in the

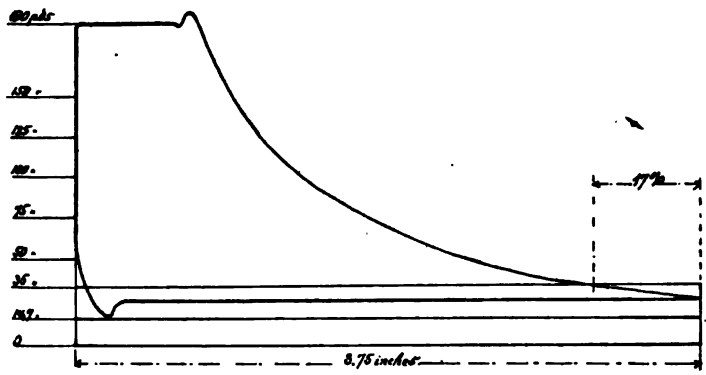


Figure 201. Suction Depression.

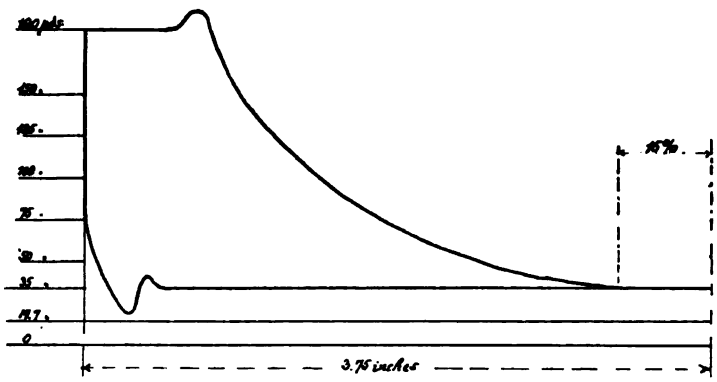


Figure 202. Suction Valve Sticking.

opening of the valve is caused, especially at the beginning of the opening and closing time. Such valves are usually noisy.

Figure 204 shows where the *suction valve or piston or both leak,* and Figure 205 shows *leaky discharge valves,* which are also indicated by the pressure being higher at the beginning of the stroke than the back pressure indicates. The actual compression curve is then higher than the adiabatic curve nearly throughout the stroke, depending, of course, on the degree of defect.

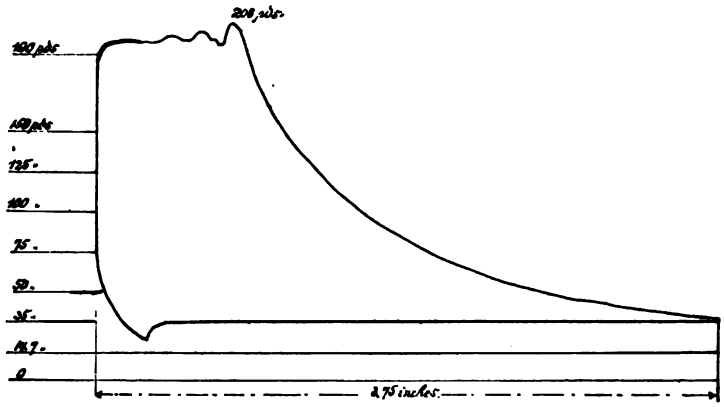


Figure 203. Excessive Final Pressure.

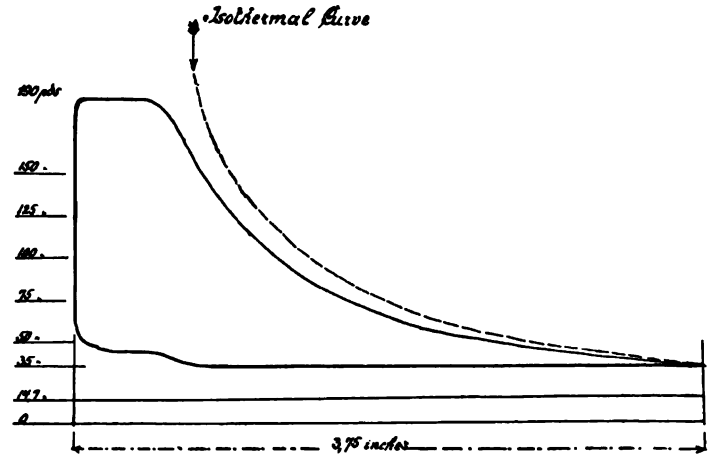


Figure 204. Leaky Suction Valve.

Figure 206 shows the *suction valve opening too slowly,* but the compressor is filled correctly.

Figure 207 shows a diagram taken from a *double-acting wet compressor* (*Linde*). This diagram is interesting in so far as it shows that wet compression does not reduce the power required for compressing ammonia gas. The actual compression curve rises above the adiabatic curve at nearly half of the stroke and then falls below.

The action of the suction and discharge valves, as well as the re-expansion, can be examined with scarcely any further

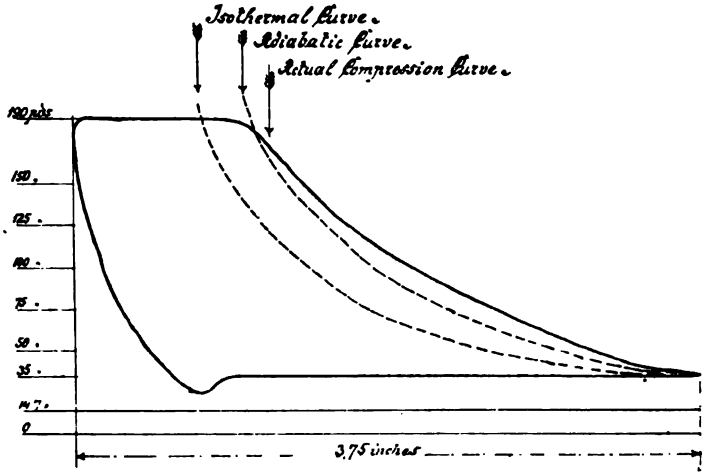


Figure 205. Leaky Discharge Valve.

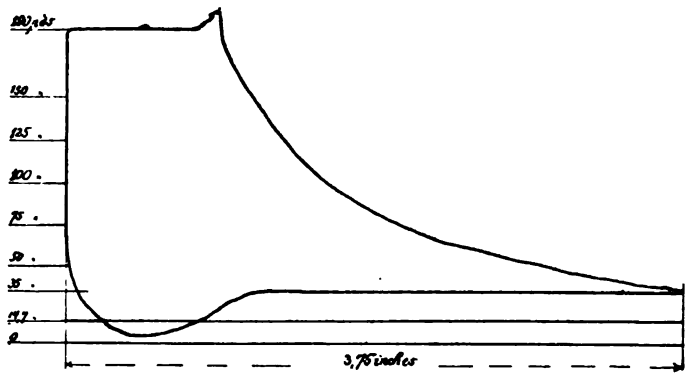


Figure 206. Suction Valve Opens too Slowly.

preparation, but for leakages and other faults, it becomes necessary to lay in the adiabatic, isothermal, vacuum, actual suction and discharge pressure lines on the diagram. The suction pressure and discharge pressure lines should be obtained from correct gauges or by a mercury column, the latter being by far the most reliable. The suction and discharge pipes should also be provided with correct thermometers, as a great many of the faults of the compressor can readily be observed by regular readings of the thermometers.

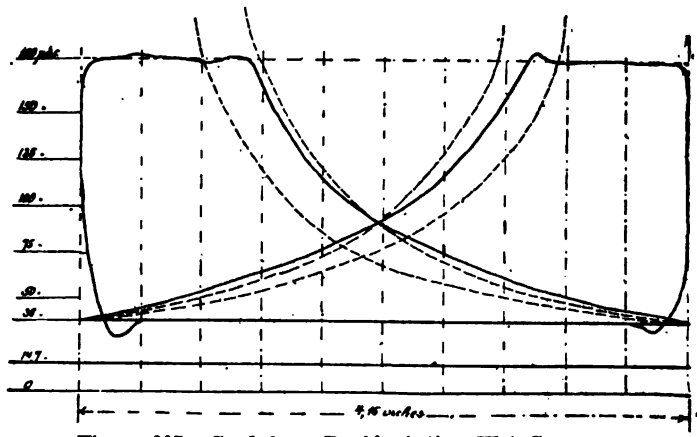


Figure 207. Card from Double-Acting Wet Compressor.

From the foregoing statements and explanations it may readily be seen that a refrigerating plant, which is properly equipped with the necessary indicating and observing instruments, can undoubtedly be brought up to the highest efficiency by a thinking engineer, who will make frequent and intelligent use of them, whereas, if simply guesswork is depended upon, poor efficiency and trouble will be a daily occurrence.

Absorption System

The absorption system, as its name indicates, is based on the fact that water absorbs ammonia gas at a low temperature and pressure and gives it off again at a high temperature and pressure. Water is able to absorb many times its volume of ammonia, the quantity of gas dissolved varying with the temperature and pressure, as shown in the table on page 569. For instance, at 32° F. and atmospheric pressure, water will absorb 1180 times its volume of ammonia, while at 212° F. and atmospheric pressure, 97 volumes of gas can be dissolved.

The ammonia gas is not merely dissolved in the water, but forms a chemical combination with it called ammonium hydrate, and as a consequence heat is generated. The quantity of heat developed is considerable, and when one pound of anhydrous ammonia gas is absorbed by a large quantity of water, it amounts to about 925 B. T. U.

Principle of Absorption Machine.

In principle, an absorption plant is the same as an ammonia compression plant, the suction and compression of the compression system corresponding respectively to the absorber and generator in the absorption system.

The absorption machine is somewhat similar to the vacuum machine, with the difference, however, that instead of water vapor being absorbed by sulphuric acid under a vacuum, the ammonia vapor is absorbed by water under pressure. It can also be compared with a condensing steam engine. The generator produces ammonia gas, the same as the steam boiler produces steam. In both cases, after the vapors have performed their work, they are condensed, the steam to water, the ammonia gas in the absorber to so-called aqua ammonia, the solution of ammonia gas in water. A pump delivers the liquid back in the gen-

Solubility of Ammonia in Water at Different Temperatures at Atmospheric Pressure.

Degrees Fahr.	Lb. of Ammonia to 1 lb. of water.	Vol. of Ammonia in 1 vol. of water.	Degrees Fahr.	Lb. of Ammonia to 1 lb. of water.	Vol. of Ammonia in 1 vol. of water.
32.0	0.899	1,180	125.6	0.274	359
35.6	0.853	1,120	129.2	0.265	348
39.2	0.809	1,062	132.8	0.256	336
42.8	0.765	1,005	136.4	0.247	324
46.4	0.724	951	140.0	0.238	312
50.0	0.684	898	143.6	0.229	301
53.6	0.646	848	147.2	0.220	289
57.2	0.611	802	150.8	0.211	277
60.8	0.578	759	154.4	0.202	265
64.4	0.546	717	158.0	0.194	254
68.0	0.518	683	161.6	0.186	244
71.6	0.490	643	165.2	0.178	234
75.2	0.467	613	168.8	0.170	223
78.8	0.446	585	172.4	0.162	212
82.4	0.426	559	176.0	0.154	202
86.0	0.408	536	179.6	0.146	192
89.2	0.393	516	183.2	0.138	181
93.2	0.378	496	186.8	0.130	170
96.8	0.363	478	190.4	0.122	160
100.4	0.350	459	194.0	0.114	149
104.0	0.338	444	197.6	0.106	139
107.6	0.326	428	201.2	0.098	128
111.2	0.315	414	204.8	0.090	118
114.8	0.303	399	208.4	0.082	107
118.4	0.294	386	212.0	0.074	97
122.0	0.284	373			

erator in a similar manner as the feed pump delivers the condenser water into the boiler.

There are several different makes of absorption machines. The following description of the operation of an absorption machine is a general one, and though each make may differ in some detail or other, in principle they are similar.

Operation of Absorption Machine.

A strong solution of ammonia in water is heated in the *generator* by means of steam passing through a coil, and the ammonia vapor is thereby separated from the water. A pressure is set up in the still by the escaping ammonia gas, and when the

Solubility of Ammonia in Water at Different Temperatures and Pressures.*

Absolute pressure in lbs. per sq. in.	32° F.		68° F.		104° F.		212° F.	
	Lbs.	Vols.	Lbs.	Vols.	Lbs.	Vols.	Lbs.	Vols.
14.67	0.899	1,180	0.518	683	0.338	443	0.074	97
15.44	0.937	1,231	0.535	703	0.349	458	0.078	102
16.41	0.980	1,287	0.556	730	0.363	476	0.083	109
17.37	1.029	1,351	0.574	754	0.378	496	0.088	115
18.34	1.077	1,414	0.594	781	0.391	513	0.092	120
19.30	1.126	1,478	0.613	805	0.404	531	0.096	126
20.27	1.177	1,546	0.632	830	0.414	543	0.101	132
21.23	1.236	1,615	0.651	855	0.425	558	0.106	139
22.19	1.283	1,685	0.669	878	0.434	570	0.110	140
23.16	1.336	1,754	0.685	894	0.445	584	0.115	151
24.13	1.388	1,823	0.704	924	0.454	596	0.120	157
25.09	1.442	1,894	0.722	948	0.463	609	0.125	164
26.06	1.496	1,965	0.741	973	0.472	619	0.130	170
27.02	1.549	2,034	0.761	999	0.479	629	0.135	177
27.99	1.603	2,105	0.780	1,023	0.486	638	....	...
28.95	1.656	2,175	0.801	1,052	0.493	647	....	...
30.88	1.758	2,309	0.842	1,106	0.511	671	....	...
32.81	1.861	2,444	0.881	1,157	0.530	696	....	...
34.74	1.966	2,582	0.919	1,207	0.547	718	....	...
36.67	2.020	2,718	0.955	1,254	0.565	742	....	...
38.60	....	....	0.992	1,302	0.579	764	....	...
40.53	....	....	....	....	0.594	780	....	...

latter then enters the condenser, consisting of a series of pipe coils, it is liquefied by the combined action of this pressure and the cooling effect of the condenser water. The resulting liquid anhydrous ammonia flows into the liquid receiver, whence it passes an expansion valve, and enters the refrigerating coils. Here it evaporates and causes refrigeration in exactly the same manner as in a compression plant. It then enters the so-called *absorber,* containing weak aqua ammonia, which has remained in the generator after the distillation of the ammonia, and which had then flown into the absorber. Here a strong aqua ammonia is again produced, which is delivered by a pump into the generator, and the cycle of operation begins anew.

* Sims.

Strength of Ammonia Liquors.

Per Cent Ammonia by Weight.	Specific Gravity.	Degrees Beaumé Water 10.	Degrees Beaumé Water 0.	Per Cent Ammonia by Weight.	Specific Gravity.	Degrees Beaumé Water 10.	Degrees Beaumé Water 0.
0	1.000	10	0	20	0.925	21.7	11.2
1	0.993	11	1	22	0.919	22.8	12.3
2	0.986	12	2	24	0.913	23.9	13.2
4	0.979	13	3	26	0.907	24.8	14.3
6	0.972	14	4	28	0.902	25.7	15.2
8	0.966	15	5	30	0.897	26.6	16.2
10	0.960	16	6	32	0.892	27.5	17.3
12	0.953	17.1	7	34	0.888	28.4	18.2
14	0.945	18.3	8.2	36	0.884	29.3	19.1
16	0.938	19.5	9.2	38	0.880	30.2	20.0
18	0.931	20.7	10.3				

The heat generated by the absorption of ammonia gas in the absorber is removed by cooling coils, because the lower the temperature, the more ammonia can be absorbed. The strong and cold liquid on its way to the generator meets the hot and weak aqua ammonia flowing to the absorber, in the so-called *heat exchanger* or *economizer,* where the weak liquid is cooled and the strong ammonia liquid heated. This heat exchanger should equalize the temperatures of the two liquids flowing in opposite directions, as much as possible, the amount of equalization depending of course, to a great extent, on the number of square feet of its surface.

After leaving the exchanger and before entering the generator, the strong liquid passes through the *analyzer,* a system of iron trays where it comes in contact with the ammonia vapors expelled from the boiling aqua ammonia in the generator.

In this manner, an increase in temperature of the strong liquid is obtained, and the ammonia vapors are deprived of some of their water.

From the analyzer, the strong liquid is discharged into the generator, while the ammonia vapor passes into the so-called *rectifier.* This device consists of cooling coils, over which water is constantly flowing. Its duty is to condense the small quantities

of water which are invariably carried along as steam by the ammonia coming from the generator. The ammonia is thus made practically anhydrous. In this water-free condition, the ammonia vapors enter the condenser as described above.

The liquid separated by the last two apparatus is collected in traps and returned to the generator.

Water can be economized in absorption machines, by using the same cooling water first on the condenser and then in the absorber.

Absorption machines are also operated by exhaust steam, when a sufficient supply is available. Most all of these machines are working very successfully.

Construction of Absorption Machines.

The construction of the absorption machine is shown in the description of two of the most common makes as given below. Though there are a number of other makes on the market, they are similar to the two described here. For a complete description of the various machines, the reader is referred to G. T. Voorhees "The Absorption Refrigerating Machine."

Carbondale Absorption Machine.—There are three types of this machine, the *shell,* the *atmospheric* and the *tubular* type. The shell type is well-suited for condensing water that is free from dirt and incrusting solids. In the case of warm or muddy water, the atmospheric type is recommended. Both of these types differ only in minor details from the tubular type. The construction of the latter type may be made clear by referring to Figure 208.

The construction of the various apparatus used in this absorption machine is as follows:

Generator.—This is made of one piece, and provided with return bend coils of extra heavy pipe. In smaller sizes, the generator shell is made of cast iron, in the larger, of welded steel.

Rectifier.—There are two types, the atmospheric type and the tubular type, both constructed on the multiple drip trap system. The tubular type is advantageous where it is desirable to conduct the waste water under pressure from the rectifier to some point at a higher level.

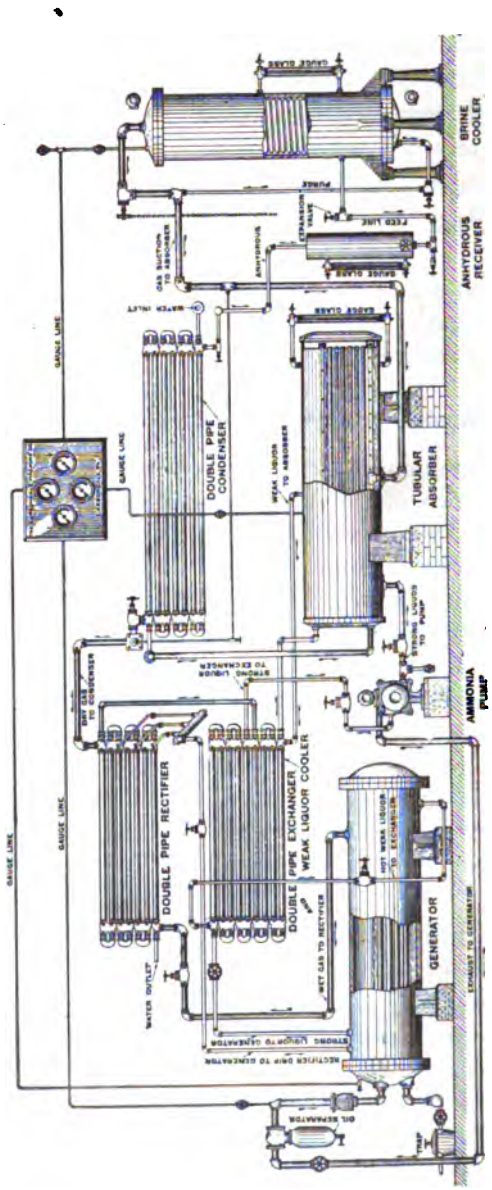


Figure 208. Carbondale Absorption Machine.

Condenser.—There are three types, the shell type, double-pipe type, and the atmospheric type. The shell type has a high water velocity in the coils. It is more difficult to clean than the double-pipe type. The atmospheric type is especially adapted for warm water or water having a great deal of sediment, as it can be readily cleaned. Where sea-water is used, this construction with galvanized pipe is recommended, and it should be kept painted to prevent corrosion.

Exchanger.—There are two types, the shell and coil type, and the double-pipe type. The former can be used either vertically or horizontally, and may be placed above the generator. In the double-pipe type, the hot weak liquor passes through the inner pipe, and the cool strong liquor in the annular space between the pipes.

Absorber.—All the types of Carbondale absorbers, viz., shell, atmospheric, double-pipe and tubular, are designed on the counter-current principle. The shell or tubular type is used where floor space is restricted. The atmospheric type is efficient where water is warm, as the evaporation of water at high temperature is rapid, and consequently a smaller quantity is required than with either the shell or tubular type.

Brine Cooler.—The Carbondale brine cooler is a vertical shell fitted with helical coils. The brine coils are submerged in liquid ammonia while the brine is rapidly circulated through the coils by a pump. The brine has a long travel through the coils and can be cooled through a large range, i. e. 10° to 15° F. if desired.

The horizontal tubular type brine coolers have as a special feature, an expansion joint which prevents leakage due to unequal expansion of shell and tubes.

Ammonia Pump.—Two types are made by the Carbondale Co., i. e., a power pump and a steam-driven pump. The power pump is of the single-cylinder, double-acting type, and is provided with a long stuffing box and fly-wheel. The steam ammonia pump is fitted with steel valves, having removable steel seats, extra long stuffing box, and as with the power pump, a drowning box. The drowning box prevents accumulation of oil in the ammonia system. The steam pump is furthermore pro-

vided with an automatic pump regulator which maintains a constant level in the absorber.

Vogt Absorption Machine.—The following paragraphs briefly describe the various apparatus which make up this machine (see Figure 209).

Generator.—This may consist of one cylinder but usually consists of two or more horizontal cylinders, one above the other. For live steam, they require 40 to 50 pounds pressure, and for exhaust steam, operate with as low as 1 pound.

Analyzer.—This is placed over the generator, and is a cylinder containing a series of perforated trays and baffle plates.

Rectifier.—This may be tubular or double-pipe or atmospheric. In the multitubular type, the strong liquid travels through the inner tubes and the gas in the outside pipe.

Condenser.—This may be of the double-pipe type, atmospheric or horizontal tubular. In the double-pipe type, the gas enters at the top and passes through the annular space between the two pipes, leaving at the bottom. The cooling water enters at the bottom, travels through the inner pipe and leaves at the top.

Weak Liquor Cooler.—This is also of the double-pipe type, and is constructed the same as the condenser.

Exchanger.—This may be either of the tubular or double-pipe type. In the multitubular type, passage is provided at each end of pipe for the liquid in the outside pipe to circulate from the top to the bottom pipe. The two liquids travel in opposite directions.

Absorber.—In the horizontal tubular type, the weak liquor enters at the top through two or more spray nozzles and the gas enters at the bottom and is distributed over the entire length of the absorber by a perforated baffle. Other types of absorbers are the vertical tubular, double-pipe, and atmospheric.

Ammonia Pump.—This pump is of the fly-wheel type, and is built in sizes ranging from 6 to 300 tons refrigeration, and runs at low speed, i. e., 25 to 30 revolutions per minute. It is regulated by a Mason governor, so that it will maintain a constant speed under varying steam pressure. The speed can be changed instantly by inserting a key and turning to the right for a lower, and to the left for a higher speed.

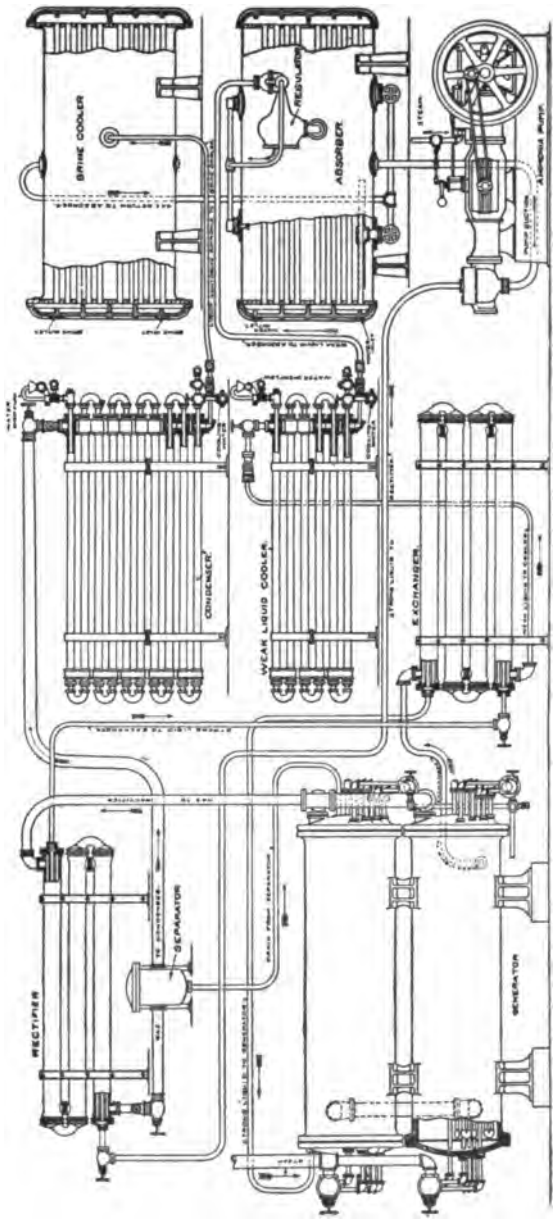


Figure 209. Vogt Absorption Machine.

Brine Cooler.—The brine cooler may be of the horizontal tubular type or of the double-pipe type. The horizontal tubular type consists of a welded steel shell flanged at each end with heavy steel heads welded thereto. The heads are drilled to receive a number of 2-inch extra heavy tubes, which are expanded in the heads. The liquid ammonia is fed to the shell of the cooler, which is usually about three-fourths full of liquid.

Cycle of Operation.

The absorption system is a combination of two cycles, sometimes called a compound cycle, which is not reversible. For the latter reason, the theoretical working condition can not be expressed as simply as in the case of a compression system, still the first law of thermodynamics can be applied, since the work done and heat generated in the first cycle must be equal to the amount of work and heat withdrawn in the second cycle of operations.

The absorption cycle can be expressed by the equation:

$$Q + Q_1 + Q_2 = H + H_1$$

Q = mechanical heat equivalent of the work required by the ammonia pump.

Q_1 = heat taken up by one pound of ammonia in the refrigerator.

Q_2 = heat given off by the steam to the ammonia in the generator.

H = heat given off by a pound of ammonia in the condenser and imparted to the cooling water.

H_1 = heat given off by one pound of ammonia in the absorber and taken up by the cooling water.

The amount of heat removed from the ammonia in the condenser is equal to the latent heat of evaporation of the ammonia at its condenser temperature, the latter being from 3° to 8° F. higher than the temperature of the cooling water leaving the condenser. The latent heat can be found from the ammonia table on page 490. For instance, if the pressure in the generator (and consequently in the condenser), is 165 pounds gauge, the latent heat of evaporation of the ammonia is found to be 489.4 B.T.U. Consequently H, the heat given off by one pound of ammonia in the condenser, is 489.4 B.T.U.

The *heat quantity* H_1 consists of the absorption heat, or the heat generated by the combining of the ammonia gas with the

weak aqua, which is designated by the letter h, and the heat which the weak liquid contains when entering the absorber, h_1. From this must be subtracted the amount of heat absorbed by the cold ammonia coming from the refrigerator, this quantity being designated by the letter h_2. The following equation, in which all heat quantities are given in B.T.U., is then obtained:

$$H_1 = h + h_1 - h_2.$$

The *quantity of heat h,* i. e. the heat generated by the absorption of the ammonia gas in the absorber, can be found by this formula:

$$h = 925 \times w - \frac{142 \times (2 \times w + w^2)}{w_1}$$

h = number of B.T.U. developed by absorption of w pounds of ammonia vapor by the weak liquid.

w = number of pounds of ammonia gas absorbed per each $1 + w_1$ pounds of weak liquid.

w_1 = number of pounds of water containing one pound of ammonia, in the weak liquid.

Example.—The weak liquid enters the absorber with a strength of 12% and leaves it with 28% of ammonia. How many heat units are developed by the absorption of the ammonia?

Solution.—The 12% ammonia contains 12 pounds of ammonia to 88 pounds of water; consequently one pound of ammonia is dissolved in $\frac{88}{12}$ or 7.33 pounds of water, or $w_1 = 7.33$, and $1 + w_1 = 8.33$.

Since a 28% solution contains 28 pounds of ammonia to 72 pounds of water, each 8.33 pounds of weak liquid must absorb $\frac{28 \times 7.33}{72} - 1 = 2.85 - 1 = 1.85$ pounds of ammonia. The absorption of this amount of ammonia will give a 28% solution. The heat developed is therefore:

$$h = 925 \times 1.85 - \frac{142 \times (1.85 \times 2 + 1.85 \times 1.85)}{7.33} = 1{,}573.7$$

B.T.U.,

or $\frac{1{,}573.7}{1.85} = 850.4$ B.T.U. per pound of ammonia absorbed by the weak liquid.

Before being able to calculate the *quantity of heat* h_1, which is the amount of heat contained in the weak liquid when entering the absorber, it is necessary first to determine the number of pounds of strong liquid which must be circulated per pound of ammonia. This can be done according to the following formula:

$$w_2 = \frac{(100 - f) \times 100}{(100 - f) \times g - (100 - g) \times f}$$

w_2 = pounds of strong liquid.
f = percentage of ammonia contained in weak liquid.
g = percentage of ammonia contained in strong liquid.

Example.—If the weak liquid enters the absorber with 12% of ammonia and leaves it with 28%, as in the preceding example, how many pounds of strong aqua ammonia must be circulated?

Solution.—

$$w_2 = \frac{(100 - 12) \times 100}{(100 - 12) \times 28 - (100 - 28) \times 12} = 5.5$$

5.5 pounds of strong liquid of 28% must be circulated. If the temperature of the weak liquid is called t, and that of the strong liquid t_1, and the specific heat of the weak liquid is designated by c, the following formula for the calculation of h_1 is obtained:

$$h_1 = (w_2 - 1) \times (t - t_1) \times c.$$

The specific heat of the liquid being very nearly 1, c may be omitted. If we assume that the difference between t and t_1 is 6°, then h_1 would be:

$$h_1 = (5.5 - 1) \times 6 = 27 \text{ B.T.U.}$$

For each pound of ammonia circulated, there are 27 B.T.U. above the absorber temperature, contained in the weak liquid, when entering the absorber.

The *quantity of heat* h_2, which is the heat absorbed by the cold ammonia coming from the refrigerator, is negative, as has already been remarked above, because the temperature of the ammonia gas coming from the refrigerator is lower than the temperature in the absorber, and consequently heat must be given off by the absorber to the ammonia gas.

The amount of heat h_2 can be calculated according to this formula:

$$h_2 = (t - t_1) \times c.$$

t = temperature in degrees F. of strong liquid leaving absorber.
t_1 = temperature in degrees F. of ammonia gas entering absorber.
c = specific heat of ammonia gas, which may be taken at 0.508.

Example.—Assuming the temperature in the absorber to be 130° F. and the ammonia gas to enter the absorber at 18 pounds gauge pressure, how much heat in B.T.U. is absorbed by a pound of cold ammonia on entering the absorber?

Solution.—The temperature corresponding to 18 pounds of pressure is 3.5° F., and we therefore have:

$$h_2 = (130 - 3.5) \times 0.508 = 64.3.$$

64.3 B.T.U. are absorbed by each pound of ammonia gas entering the absorber.

The mechanical heat equivalent of the work required by the ammonia pump, designated in previous paragraphs by the letter Q, is comparatively small, and may be neglected in the calculation, the result being still sufficiently accurate for practical purposes.

If desired, however, it can be found in B.T.U., according to the following formula:

$$Q = \frac{w_2 \times (h - h_1)}{s \times 778}$$

w_2 = number of pounds of strong liquid pumped for every pound of ammonia.
h = head or height in feet of the strong liquid corresponding to its pressure.
h_1 = head in feet of the weak liquid corresponding to pressure.
s = specific gravity of strong liquid.

Example.—The pressure in the absorber is 18 pounds gauge, in the generator 165 pounds, and 5.5 pounds of strong liquid are circulated per pound of ammonia. The temperature in the absorber is 130° F. The specific gravity of the strong solution is 0.902. How many B.T.U. are expended as work in the ammonia pump?

Solution.—First, the difference between the two heads, $h - h_1$ is found by taking the difference between the two pressures, multiplying this by 144 square inches, the base of one cubic foot, and dividing the result by 61.55, the weight of one cubic foot of water at 130° F.

$$h - h_1 = \frac{(165 - 18) \times 144}{61.55} = 343.9.$$

Now Q can be calculated:

$$Q = \frac{5.5 \times 343.9}{0.902 \times 778} = 2.70 \text{ B.T.U.}$$

The friction in these pumps averages about 25%.

If this friction is assumed in the above example, Q is then $\frac{2.70}{0.75}$ or 3.6 B.T.U.

It can readily be seen that the energy required to drive the pump is comparatively small, and may therefore be neglected. In this case, the friction loss is the only one taken into account.

The *heat quantity* Q_1, i. e. the amount of heat absorbed by the ammonia in the refrigerating room, can be found as follows:

$$Q_1 = H - (t - t_1) \times c.$$

Q_1 = B.T.U. absorbed by one pound of ammonia.
H = total latent heat of vaporization in B.T.U. at the temperature t_1.
t = temperature of the liquid ammonia in degrees F. before passing the expansion valve.
t_1 = temperature in degrees F. after passing expansion valve.
c = specific heat of the liquid, which may be taken as 1.1.

Example.—How many heat units are absorbed in the refrigerator by one pound of ammonia, if the pressure in the absorber is 18 pounds gauge, and in the generator 165 pounds?

Solution.—The values for H, t and t_1 can be taken from the ammonia table and inserted in the above formula, which then reads:

$$Q_1 = 567.2 - (89.5 - 3.5) \times 1.1 = 472.6.$$

472.6 B.T.U. are absorbed by one pound of ammonia in the refrigerator.

From the data calculated in the preceding paragraphs, we can now find the *amount of heat* Q_2 given off by the steam in the generator and used to expel one pound of ammonia gas.

It will be remembered that the cycle of operation was expressed by the following equation:

$$Q + Q_1 + Q_2 = H + H_1.$$

From this it follows that:

$$Q_2 = H + H_1 - Q - Q_1.$$

In the ammonia table it has been found that H = 489.4 (see page 576), and by calculation the values for H_1, Q and Q_1 have been found as follows:

$H_1 = 850.4 + 27 - 64.3 = 813.1$ B. T. U.

$Q = 3.6$ B.T.U.

$Q_1 = 472.6$ B.T.U.

Consequently:

$$Q_2 = 489.4 + 850.4 - 3.6 - 472.6 = 863.6 \text{ B.T.U.}$$

From this it will be perceived that 863.6 B.T.U. must be furnished by the steam in the generator for each pound of ammonia which is expelled from the aqua ammonia and circulated in the system. This, however, does not take into consideration losses by radiation, conduction, defects in construction and operation of the apparatus, etc. In reality this loss is considerable, and amounts in practice to from 35% to 50% in well conducted plants. The actual heat consumption under average conditions would, therefore, probably be not less than $\frac{863.6}{0.65}$ or 1,329 B.T.U. per pound of ammonia circulated, and usually considerably more, the losses being generally closer to 45% than to 35%.

Amount of Cooling Water Required.

The amount of water required for an absorption plant for best economy is larger than that necessary for a compression machine.

The quantity of water to be employed per pound of ammonia circulated can be calculated by the following formula:

$$P = \frac{H + H_1}{t - t_1}$$

P = number of pounds of water.

H = heat in B.T.U. absorbed by the cooling water from one pound of ammonia in the condenser.

H_1 = heat in B.T.U. absorbed by the cooling water from one pound of ammonia in the absorber.

t = average temperature in degrees F. of cooling water leaving both vessels.

t_1 = initial temperature in degrees F. of the cooling water when entering condenser or absorber.

Example.—How many pounds of cooling water are required per pound of ammonia circulated, under the conditions of the preceding examples, if the initial temperature of the cooling water is 60° F., and the final temperature 80° F?

Solution.—In previous examples, the cooling water was found to absorb 489.4 B.T.U. in the condenser and 813.1 B.T.U. in the absorber. Therefore, we have:

$$P = \frac{489.4 + 813.1}{80 - 60} = 65.12.$$

65.12 pounds of cooling water must be employed per pound of ammonia.

Knowing this, it is a comparatively easy matter to find the amount of cooling water required per minute for a certain number of tons of refrigeration in 24 hours. It may be done according to the following formula:

$$P_1 = \frac{(H + H_1) \times n \times 288{,}000}{(t - t_1) \times Q_1 \times 1{,}440}$$

P_1 = pounds of water per minute.
H = heat in B. T. U. absorbed by one pound of ammonia in condenser.
H_1 = heat in B.T.U. absorbed by one pound of ammonia in absorber.
n = number of tons of refrigeration in 24 hours.
t and t_1 = final and initial temperatures in degrees F. of cooling water for condenser and absorber.
Q_1 = B.T.U. absorbed by one pound of ammonia in refrigerator.
1,440 = number of minutes in 24 hours.
288,000 = number of B.T.U. in one ton of refrigeration.

Example.—How many pounds of cooling water are required per minute for a 10-ton absorption machine under the conditions of the preceding examples?

Solution.—$P_1 = \frac{(489.4 + 813.1) \times 10 \times 288{,}000}{(80 - 60) \times 472.6 \times 1{,}440} = 275.6.$

Theoretically 275.6 pounds of water would be required, heat transmission being perfect. In practice this is not the case, however, and on the average about 25% more is required.

If the water is expensive, economizing may be done by using the water first over the condenser and then in the absorber,

which naturally increases the temperature of the liquid leaving the absorber correspondingly. Water can also be economized as described under cooling towers (see page 529).

Horse-power Required to Drive Pump.

In order to be able to find the horse-power necessary for driving the ammonia pump, the following calculation must be made.

The number of pounds of strong ammonia liquid required per minute for a certain number of tons of refrigeration in 24 hours is multiplied by the weight of the solution, and by the head in feet corresponding to the pressures in generator and absorber. The result is divided by the specific gravity of the strong liquid multiplied by 33,000.

Expressed in a formula, it is as follows:

$$\text{H. P.} = \frac{n \times 288{,}000 \times w \times (p - p_1) \times 144}{s \times Q_1 \times 33{,}000 \times g \times 1{,}440}$$

n = number of tons of refrigeration in 24 hours.
w = pounds of strong liquid circulated per pound of ammonia.
p = pressure in generator, in pounds per square inch.
p_1 = pressure in absorber, in pounds per square inch.
s = specific gravity of strong solution.
Q_1 = heat in B.T.U. absorbed by one pound of ammonia in refrigerator.
g = weight in pounds of one cubic foot of water at temperature of absorber.

Example.—By using the data of the preceding example, we have:

$$\text{H. P.} = \frac{10 \times 288{,}000 \times 5.5 \times (165 - 18) \times 144}{0.902 \times 472.6 \times 33{,}000 \times 61.653 \times 1{,}440} = 0.268.$$

0.268 H. P. would be necessary to drive the pump, if there were no friction. In reality, however, there exists a friction of about 25%, so that the horse-power actually required in the above case would amount to $\frac{0.268}{0.75}$ or 0.357.

Efficiency of Absorption Plant.

By the efficiency of an absorption plant is generally understood that percentage of the total heat furnished by the steam

to the generator and ammonia pump which is utilized for refrigerating. The theoretical maximum efficiency can be expressed by the formula:

$$E = \frac{Q_1 \times 100}{Q + Q_2}.$$

E = theoretical maximum efficiency in per cent.
Q_1 = heat in B.T.U. absorbed by one pound of ammonia in the refrigerator.
Q = number of B.T.U. furnished to the generator.
Q_2 = number of B.T.U. furnished to the pump.

Example.—What is the theoretical maximum efficiency of a 10-ton absorption plant under the conditions of the preceding examples?

Solution.— $$E = \frac{472.6 \times 100}{3.6 + 863.6} = 54.5\%.$$

The theoretical maximum efficiency is 54.5%. There are, however, losses by radiation, conduction, etc., which must be taken into account if the actual efficiency is to be found. These losses amount to at least 35%, so that in the above case under these conditions the actual efficiency would be:

$$E_1 = \frac{472.6 \times 100 \times 0.65}{3.6 + 863.6} = 35.4\%.$$

Calculation of the efficiency of the plant can also be based on the heat furnished by the coal, i. e. the refrigerating effect of one pound of ammonia can be compared with the number of B.T.U. furnished by one pound of the fuel. If we assume the efficiency of the steam boiler to be 65%, then the actual efficiency of the absorption plant as compared with the number of B.T.U. derived from the coal would be:

$$E_2 = \frac{472.6 \times 100 \times 0.65 \times 0.65}{3.6 + 863.6} = 23.0\%.$$

This means that every 1 B.T.U. contained in the coal, 0.230 B.T.U. is abstracted from the refrigerating room, or for every 4.35 B.T.U. in the coal, 1 B.T.U. can be obtained for refrigeration. This agrees very closely with results obtained in practice. A practical test made on an absorption machine showed that of 27,000,000 B.T.U. contained in the coal used during 24

hours, 6,218,816 B.T.U. were obtained for refrigeration, which is equal to an actual efficiency of $\frac{6{,}218{,}816 \times 100}{27{,}000{,}000}$ or 23%.

Losses in Absorption System.

If the exchanger, or economizer, as it is sometimes called, permitted a complete exchange of heat, which of course is a condition never met with in practice, then all the heat given off by the steam in the generator coils less the portion lost by radiation and conduction, would be found in the vaporized ammonia, and consequently in the water coming from the condenser. Under such ideal conditions, the total heat expended, besides including the heat lost by radiation and conduction, would be the heat required for vaporizing the ammonia and some of the water; a small quantity of water in the form of steam being unavoidably carried along with the ammonia vapors to the rectifier and condenser.

The liquid anhydrous ammonia, in becoming vaporized in the refrigerator, takes up as much heat as was given off during its cooling and liquefaction in the condenser, less the amount used for cooling liquid from the temperature of the condenser, to the temperature of the refrigerator. The total heat less the small quantity mentioned above, is taken up by the cooling water in the absorber when the ammonia vapors are dissolved in the weak aqua ammonia. If, therefore, the amount of heat taken up by the ammonia in the refrigerator together with that given off by the steam in the generator were equal to the heat carried away by he cooling water from the condenser plus that abstracted from the absorber, then the sources of loss would be:

1. Inefficiency of the economizer.

2. Radiation and conduction from all vessels and pipes that are above normal temperature (temperature of surrounding air).

3. Useless evaporation of water which passes into the rectifier and condenser.

4. Conduction of heat into all vessels and pipes that are below normal temperature.

5. Water passing into the refrigerator along with the liquid ammonia.

Comparison Between Absorption and Compression Systems.

In the absorption system, the steam must furnish considerably more heat than in the compression system. This is mainly due to the fact that in the absorption system the heat of vaporization taken up by the anhydrous ammonia in the refrigerating room is given off in the absorber, so that the entire heat of vaporization required to produce the ammonia vapor in the generator has to be supplied by the steam. In other words, the ammonia in an absorption machine is gasified twice during one complete cycle and must necessarily be liquefied twice, viz., once in the condenser and once in the absorber. The amount of water required must, therefore, be larger than if only one liquefaction took place.

In the compression system, the ammonia vapor passes directly from the refrigerator to the compressor, and power must be expended merely in increasing the pressure. This also raises the temperature of the ammonia accordingly, and enables liquefaction at ordinary temperatures, i. e. gives the cooling water, which is of ordinary temperature and therefore warmer than the ammonia vapor coming from the refrigerator, a chance to remove the heat of vaporization from the ammonia and to condense the vapor to a liquid.

However, in general, the efficiencies of the absorption machine and of the compression machine are about the same. When a compression machine operates with high suction pressure and low discharge pressure, its efficiency is somewhat greater than that of the absorption machine. On the other hand, with low suction pressure and high discharge pressure, the efficiency of the absorption machine will be greater than that of the compression machine.

Where a *compression machine is combined with an absorption machine,* and where the exhaust steam from the steam cylinder of the compression machine is used in the absorption machine, the efficiency of the entire plant is very high.

Since a steam engine can only use the small amount of sensible heat in the steam, while the absorption machine can utilize the latent heat of the steam, it is evident that, by combining the two machines as stated above, the efficiency must be increased. Where the steam is not used in this manner, only

a very small amount of the latent heat is regained by heating the feed-water, unless it be made useful for other heating purposes.

Testing Tightness of System.

The testing of an absorption system as to tightness of pipes, joints, etc., is generally done by first applying hydrostatic pressure of 300 pounds per square inch. After the water has been drained off, a partial vacuum is formed in the system, the air being removed either by the ammonia pump or another suitable pump, or sometimes by the use of steam. In the latter case all valves are opened and steam is admitted to the coils. When the steam has forced out all the air, the valve leading to the atmosphere is closed and the steam allowed to condense, thus creating a vacuum in the system. Since the pipe joints are liable to become damaged by the heat, the use of a pump for generating the vacuum is preferable.

Charging the System.

After a vacuum has been generated in the system according to either of the methods mentioned, aqua ammonia of 26° Beaumé, containing 29.5% ammonia, is admitted at the bottom of the still until the partial vacuum is destroyed, the rest of the charge being pumped in by the ammonia pump. The ammonia drums should be placed higher than the pump, in order to prevent the latter from being gas bound. Care must be taken not to overcharge the generator when cold, as the liquid increases considerably in volume when steam is turned on to the coils, causing it to foam or boil over. On the other hand, the concentrated liquid should never be allowed to fall below the top of the steam coils.

Starting the Plant.

When almost the entire charge is in the system, a small amount of steam is admitted to the generator or still and the pressure thereby raised to about 100 pounds; it is then shut off and freely used on the rectifier coils, to cause the air remaining in the same to rise to the top, where it is drawn off through a purge cock, on the top of the coils, by means of a hose leading from the cock into a pail of water. Bubbles com-

ing out of the hose indicate that there is still air in the system; when they cease coming, the purge cock should be closed. The weak aqua ammonia line leading from the generator to absorber, as well as the gas line leading to the absorber, is then opened.

The water is now turned on for the condenser and the absorber. The latter is being filled, and since the generator is deprived of this quantity of ammonia, a corresponding amount must be recharged with the pump.

When the gauge glass of the liquid receiver shows sufficient ammonia, the expansion valves are opened, and the ammonia pump started, to deliver the enriched liquor through the exchanger into the generator. The weak liquor valve is now regulated until the gauge glass of the absorber indicates that the latter is about half-filled.

If a sufficient amount of water circulates through the absorber so that the latter remains cool, the machine is in perfect working order.

The supply of water used over the coils in the rectifier must be well regulated. If too much water is used, the ammonia gas is partly liquefied, in which state it will flow into the generator, whereby a waste of heat is entailed. If, however, too little water is employed, the water vapors which are carried along by the ammonia and which are supposed to be condensed in the rectifier, will pass into the condenser, causing a decrease in efficiency of the ammonia. It is good practice to have the coils warm from the bottom to about the center coil.

Care should be taken that the weak aqua ammonia is not too weak; if it shows less than 23° Beaumé, anhydrous ammonia must be added in order to bring it to 23°. The liquid must therefore be weighed from time to time. It is also necessary to test the system once in a while as to the possible presence of air, especially if the system is new.

The coils in the still should always be covered with liquid, as otherwise corrosion and pitting will soon destroy them. The latter two are the most dangerous enemies of an absorption plant, and they should therefore be guarded against with the greatest precaution.

The Brine System

In the direct expansion system, cooling is effected by the evaporation of ammonia in the coils with which the refrigerating rooms are equipped. In the brine system, however, the ammonia is used for cooling a salt solution, so-called brine, in a suitable tank or double-pipe cooler, and this cold brine is then circulated through a system of refrigerating coils, which are arranged in the same manner as the ammonia coils.

Preparation of Brine.

For the production of brine, common salt, calcium chloride or magnesium chloride may be employed, in nearly all cases, one of the first two. The solutions of these substances can be used for refrigerating purposes, because the freezing point lies below that of water, and the temperature can, therefore, be decreased without their becoming solid. The freezing point of brine is the lower the more concentrated the solution, and depends on the nature of the substance used. Calcium chloride brine in strong solutions freezes at a much lower temperature than ordinary salt brine of the same strength.

In order to make brine properly, the lowest temperature which the brine is to possess must be known, because on this depends the kind and quantity of substance to be used. Furthermore, a table giving the properties of brine, and a salometer or special hydrometer for determining the density of brine must be at hand.

Common Salt.

When brine is to be made from common salt, a very simple device is sufficient to properly dissolve the salt, and at the same time prevent impurities from being carried into the brine tank.

A barrel or cask is provided with a false bottom about 6 to 8 inches above the real bottom, the false bottom being made of

wooden strips one inch thick and one inch wide and placed about one-half inch apart. This is to be supported by two pieces of wood, each about six or eight inches wide, put on edge and nailed to the bottom. To permit a free circulation of water, the supports should be perforated. A piece of burlap is stretched across the top of the false bottom and nailed to the sides of the barrel.

The water inlet, to which leads a 1¼-inch supply pipe, is between the false bottom and the real bottom; the outlet for the brine is near the top of the barrel and is connected to a 1½-inch pipe. It should be provided with a strainer to prevent pieces of wood and other foreign matter from being carried along with the brine.

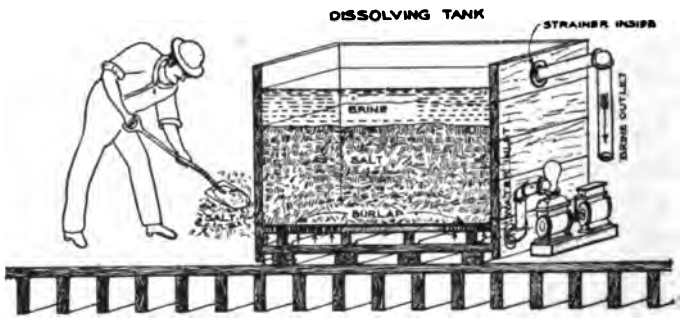


Figure 210. Filling Brine Tank with Salt.

The barrel is filled with salt and the water turned on. Entering at the bottom the water rises in the barrel, and on its way through the salt dissolves it. In proportion to the quantity dissolved, more salt is shoveled into the barrel, so that it is continually kept filled until the total calculated amount of salt has been used. No stirring is necessary. The waste matter which rises to the top should be skimmed off. If one barrel proves insufficient, more should be set up.

In order to have the brine perfectly free from impurities, it is best to filter it, which can easily be done by allowing it to flow through a small tank about three-fourths filled with excelsior or hay, and placed above the brine tank. The filtering material is covered with a piece of burlap or some other porous material. The brine enters this filter tank at the bottom and leaves it at the top.

It has proven beneficial to add to the salt about 2% of *sodium carbonate* to counteract the destructive effect of salt brine on iron.

If calcium chloride is employed, however, this should be omitted, as otherwise solid calcium carbonate would be precipitated and deposited. Another substance must be added in this case, as will be seen under calcium chloride.

The following table gives the properties of common salt brine:

Properties of Sodium Chloride Brine.

Per Cent Pure Salt By Weight	Degrees Beaumé at 60° F.	Degrees Salinometer at 60° F.	Specific Gravity at 60° F.	Pounds Salt per Gallon of Solution	Pounds Salt per Cubic Foot of Solution	Specific Heat	Freezing Point Degrees F.
1	1	4	1.007	0.084	0.63	0.992	+30.5
2	2	8	1.015	0.169	1.26	0.984	+29.3
2.5	3	10	1.019	0.212	1.58	0.980	+28.6
3	3.5	12	1.023	0.256	1.92	0.976	+27.8
3.5	4	14	1.026	0.300	2.24	0.972	+27.1
4	4.5	16	1.030	0.344	2.57	0.968	+26.6
5	5.5	20	1.037	0.433	3.24	0.960	+25.2
6	6.5	24	1.045	0.523	3.92	0.946	+23.9
7	7.6	28	1.053	0.617	4.63	0.932	+22.5
8	8.7	32	1.061	0.708	5.30	0.919	+21.2
9	9.7	36	1.068	0.802	6.00	0.905	+19.9
10	10.7	40	1.076	0.897	6.70	0.892	+18.7
12	12.6	48	1.091	1.092	8.20	0.874	+16.0
15	15.7	60	1.115	1.389	10.40	0.855	+12.2
20	20.4	80	1.155	1.928	14.40	0.829	+ 6.1
24	24	96	1.187	2.376	17.78	0.795	+ 1.2
25	25	100	1.196	2.488	18.68	0.783	+ 0.5
26	25.8	...	1.204	2.610	19.50	0.771	— 1.1
29	...	...	....	....	....	....	— 4.7

Calcium Chloride.

Calcium chloride possesses a number of properties which render it superior to common salt for the production of brine. The very low freezing point of its strong solution, which is considerably lower than that of common salt brine, makes its use indispensable where very low temperatures must be produced. In cold storage rooms, for instance, where the temperature should be below 0° F., only calcium chloride brine can be employed. It should also be given the preference where double-pipe coolers are in use, for the reason that common salt brine may freeze to a solid mass and damage the cooler if the suction pressure is

low, as may happen at times. The use of calcium chloride brine avoids this difficulty absolutely. It also appears to cause less friction when circulated than salt brine.

Its influence on metals is less pronounced than that of common salt, but it is nevertheless advisable to add about ½% of *caustic soda* to this brine.

The following table contains the properties of calcium chloride brine:

Properties of Calcium Chloride Brine.

Per Cent Pure Salt By Weight	Degrees Beaumé at 60° F.	Degrees Salinometer at 60° F.	Specific Gravity at 60° F.	Pounds Commercial Salt per Gallon of Solution	Pounds Commercial Salt per Cubic Foot of Solution	Specific Heat	Freezing Point Degrees F.
3	3	12	1.024	½	3¾	0.980	+29
5	6	27	1.041	1	7½	0.964	+27
7	9	36	1.058	1¼	8½	0.936	+25
9	10	40	1.076	1½	11¼	0.911	+23
10	11	44	1.085	1¾	13	0.896	+21
12	13	52	1.103	2	15	0.884	+18
14	15	62	1.121	2¼	17	0.868	+14
18	20	80	1.159	2½	19	0.844	+ 4
20	22	88	1.179	3	22½	0.834	— 1.5
22	24	95	1.199	3½	26	0.817	— 8
24	26	104	1.219	4	30	0.799	—17
26	28	112	1.240	4½	34	0.778	—27
28	30	120	1.261	5	37½	....	—41
30	32	...	1.283	5½	41	....	—54

It will be perceived from the table that a strong solution of calcium chloride has a much lower freezing point than a weaker one. When making calcium chloride brine, the solution should not be made stronger than necessary; a 25% solution is strong enough for all purposes, and where suction pressures above 15 pounds are carried, a 20% solution is sufficient.

In determining the strength of calcium chloride brine, as well as common salt, a salometer, or Beaumé hydrometer may be employed.

Calcium chloride is sold for refrigerating purposes in one solid piece, covered air-tight by thin sheet iron. When brine is to be made, this should first be pounded with a sledge hammer

in order to break the block into a number of smaller lumps without scattering them all over the floor. The iron shell can then be removed by means of an ax or cold chisel. A clear floor space should be selected to do this, where nothing can be injured by the moisture which rapidly collects on every piece of calcium chloride exposed to the air.

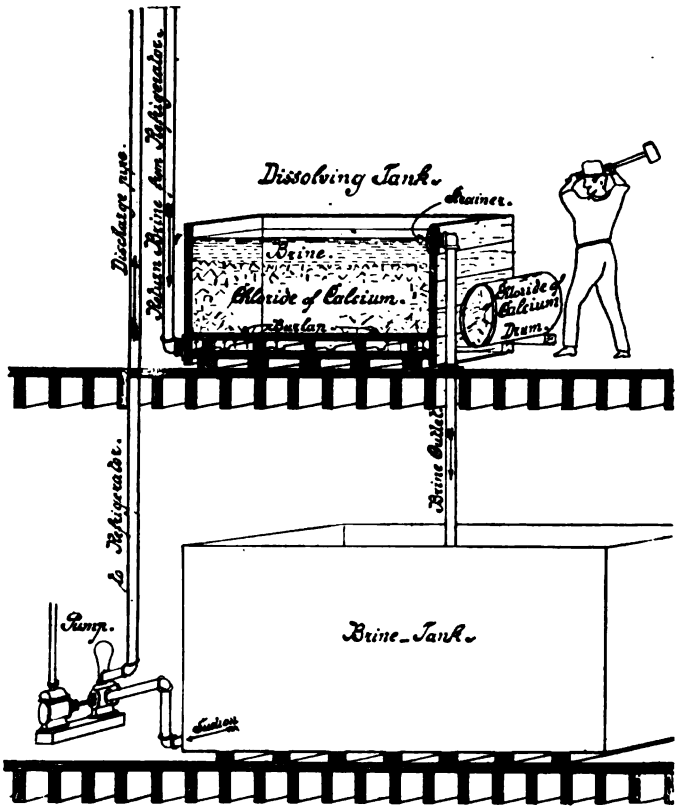


Figure 211. Filling Brine Tank with Calcium Chloride.

The easiest way to dissolve the calcium chloride lumps is to shovel them into an auxiliary tank connected to the brine pump and the brine tank. The pump sucks from the auxiliary tank and discharges into the brine tank. The water or weak brine flows from the latter into the smaller tank and over the calcium chloride, which is thereby gradually dissolved.

Since it absorbs moisture rapidly when exposed to the air, calcium chloride is sometimes employed for *drying* the air in cold storage rooms. The resulting liquid, which is strong calcium chloride brine, can be collected and added to the other brine in the tank.

Mixture of Common Salt and Calcium Chloride.

In a number of cases where common salt brine had been in use, and the introduction of calcium chloride brine seemed advantageous, calcium chloride has been added to the salt brine already in use. As far as the author is informed, the results obtained have been very satisfactory, this mixed brine having proven superior in some respects to the common salt brine.

Magnesium Chloride Brine.

Magnesium chloride is a substance whose properties are very similar to those of calcium chloride. It is even better adapted to the preparation of brine, especially where very low temperatures are to be reached, because the freezing point of its solution is lower than that of calcium chloride, and it has the further advantage that it is absolutely neutral towards iron. It is, however, more expensive, and so far is comparatively little used in this country.

Properties of Magnesium Chloride Brine.

Per Cent Water-free Salt	Specific Gravity 15° C.	Specific Heat	Ammonia Gauge Pressure lbs. per sq. in.	Freezing Point Degrees F.
0	1.00	1.00	47	32
.3	1.02	.965	43	29
5	1.04	.93	39	25.7
7	1.06	.90	35	22.1
9.5	1.08	.87	31	18.1
11.5	1.10	.84	27	14.0
14	1.12	.815	22½	9.1
16	1.14	.79	18	3.9
18	1.16	.77	13½	—2.2
20	1.18	.75	10	—7.6
22.5	1.20	.73	6	—14.8

Brine Tank.

Cooling of the brine is generally done in a tank equipped with a sufficient number of ammonia coils, through which the

liquid ammonia circulates, evaporates and then returns in a gaseous condition to the compressor. The heat necessary to evaporate the liquid ammonia is furnished by the brine, whose temperature is thereby reduced. Deprived of its heat, the brine is drawn away by the circulating pump, forced through the coils where it abstracts heat from the surrounding air, and returned in a warmer condition to the tank which it enters at the top.

Construction and Dimensions of Tank.—Brine tanks are generally made of ¼-inch or 5/16-inch tank iron, and will last a long time, provided they are butt-strap riveted and well-braced, so that the weight of the brine does not cause the walls of the tank to bulge, and provided they are well protected against corrosion. Internal bracing is better than external bracing.

Recently, brine tanks have been constructed of re-enforced concrete with very good success. The life of such a tank is naturally much longer than that of an iron tank, which is subject to corrosion.

The volume of the brine tank is in direct proportion to the refrigeration which is to be produced, and for each ton of refrigeration from 60 to 75 cubic feet of volume should be allowed. The space occupied by the ammonia coils amounts to about 3% of the volume of the tank.

Example.—What must be the dimensions of a brine tank for a capacity of 50 tons of refrigeration?

Solution.—Allowing 75 cubic feet per ton, the volume of the tank would have to be 50×75, or 3,750 cubic feet. A tank 20 feet long, 15 feet wide and 12 feet high would approximately have the required size.

Loss by Radiation.—With medium insulation, the radiation in brine tanks, if located in hot engine rooms, amounts to about 2% of the refrigeration produced. It may be said here that exceptional care should be taken in insulating a brine tank, and most decidedly so if the tank is to stand in a hot place, as for instance in or above the engine room, as is often the case.

Arrangement of Coils.—The submerged coils are fed by expansion valves either at the bottom or top, the ammonia leaving either at the top or bottom. According to the author's experience, the best results are obtained if the ammonia enters at the bottom and leaves the coils at the top.

Proper Length of Pipe Coils.—For each ton of refrigeration about 85 square feet of pipe surface should be allowed, which corresponds to:

85 × 1.611 or 137 running feet of 2-inch pipe.

85 × 2.201 or 187 running feet of 1½-inch pipe.

85 × 2.301 or 195 running feet of 1¼-inch pipe.

85 × 2.903 or 246 running feet of 1-inch pipe.

Example.—With how many feet of 1¼-inch pipe must a brine tank be equipped, if it is intended for the production of 50 tons of refrigeration?

Solution.—The brine tank must be equipped with 50 × 195 or 9,750 feet of 1¼-inch pipe.

The foregoing figures are based on an average brine temperature of 18 to 20° F. and a suction pressure of 20 to 25 pounds.

Tests on Brine Tank.

Test Number	1	2	3	4	5	6	Average
Initial temperature of brine, degrees F.....	29	26.5	22.1	23	24.5	23.7	24.8
Final temperature of brine, degrees F.....	26	22.5	18.5	19.6	20.6	19	21.0
Cooled down, degrees F..	3	4	3.6	3.4	3.9	4.7	3.76
Ammonia suction pressure, lbs.	26	26.4	25.6	25.9	26	26.2	26.1
Ammonia temperature, degrees F.	+13	+13	+13	+13	+13	+13	+13
Gallons of brine per minute	350	350	350	350	350	350	350
Hours brine tank operates	24	24	24	24	24	24	24
Tons of refrigeration per 24 hours	43.2	57.6	51.84	48	56	68	54.01
Velocity of brine in feet per minute	1.1	1.1	1.1	1.1	1.1	1.1	1.1
Square feet cooling surface	2300	2300	2300	2300	2300	2300	2300
B. T. U. per square foot per hour	223	300	270	252	289	350	280.66
B.T.U. per 1° F. difference per hour per sq. ft.....	15.5	26.3	38.0	30.7	30.7	43.2	30.7
Mean temperature difference, degrees F.	14.4	11.4	7.1	8.2	9.4	8.1	9.8
Lowest brine temperature in tank after brine cooler and pump stopped, ice machine working on tank alone, degrees F.	0	—1	—1	—1	—1	—1	—1
Cooled down, degrees F..	26	23.5	19.5	20.5	21.5	20	22

Heat Transmission.—For ordinary brine tanks which have no mechanical circulation and in which the ammonia pipes are fed by the force feed system, the heat transmission between the brine and the ammonia per square foot of surface per hour per degree F. difference in temperature is about 30 B. T. U. For the flooded system, the coefficient is about 35 B. T. U.

When mechanical circulation is used, the coefficient is about 50 B. T. U. with the force feed system, and about 60 B. T. U. with the flooded system.

On page 596 are given the results of six tests made on a brine tank by G. J. Patitz. It will be noted that the heat transmission is only about 30 B. T. U. per square foot per hour per degree F. temperature difference, as compared to 90 B. T. U. in the case of the brine cooler (see page 598).

Brine Circulating Pump.

The proper size of the pump for circulating the brine can be determined on the basis that the brine should not travel through the pipe faster than 250 feet per minute, and at the rate of 5 gallons of brine per minute per ton of refrigeration. This usually brings the brine back to the tank with a temperature of about 4° F. higher than when it left the tank.

The pump, which ought to be brass fitted, should be provided with suction and discharge stop valves. A strainer should be put in the suction pipe, so arranged by means of a by-pass and valves that it can be taken out while the pump is running. In the suction pipe a thermometer should be inserted.

The return pipe should be placed in such a manner that good circulation takes place in the tank. It should enter at the corner of the tank opposite to where the suction is located, which latter should be near the bottom.

Arrangement of Coils in Cellar.

In order to equalize the pressure in each coil in the different stories, the brine mains supplying the coils in the cellar should be so arranged, that the brine leaving each cellar coil rises to a point higher than the coil in the highest cellar.

The top of the return pipe should be provided with a vacuum breaker to prevent the brine being siphoned out.

The pipe headers in the cellars should be of ample size to

accommodate the quantity of brine delivered by the discharge pipe, the size of which is determined by the discharge opening of the pump, which should never be reduced.

Each coil of pipes connected to the discharge header in the cellars should be provided with a good valve of the size of the coil, and no more than 600 feet of 1¼-inch pipe, and no more than 1,200 feet of 2-inch pipe should be connected to one valve. Every coil should be equipped with an air vent.

Double-Pipe Brine Coolers.

Double-pipe brine coolers are constructed in practically the same manner as double-pipe condensers. They are extremely practical and efficient, being easily accessible and consequently easily kept in good condition without interfering with the brine in the tank. Since there are no pipes in the tank, the latter may be given any desired shape, the round form being generally preferred, and as the cooling of the brine is done outside of the tank, the latter merely serves as a storage tank.

The brine cooler, which is arranged in such a manner that the brine and ammonia travel in opposite directions, can be placed

Tests on Brine Cooler (4 Coils).

Test Number	1	2	3	4	5	6	Average
Brine circulated per minute, gallons	350	350	350	350	350	350	350
Initial temperature of brine, degrees F.....	32.8	30.4	25.1	27	28.1	27.1	28.4
Final temperature of brine, degrees F.	29	26.5	22.1	23	24.5	23.7	24.8
Cooled down, degrees F...	3.8	3.9	3.2	4	3.6	3.4	3.65
Ammonia temperature, degrees F.	+13	+13	+12	+13	+13	+13	+12.80
Ammonia pressure in lbs.	26	26.4	25.6	25.9	26	26.2	26.01
Cooling surface, square ft.	512	512	512	512	512	512	512
B. T. U. per hour per square foot	1274	1313	1004	1340	1206	1104	1213
Mean temperature difference, degrees F.....	17.8	15.4	11.5	11.9	13.2	12.3	13.7
Velocity of brine, feet per minute	580	580	580	580	580	580	580
B.T.U. per 1° F. difference per hour per sq. ft.....	71.6	85.3	87.3	112.6	91.4	89.8	89.7
Tons refrigeration per 24 hours	55.1	58.5	43.5	58	52.2	49.3	52.8
Capacity per single section in tons refrigeration	13.8	14.6	11	14.5	13	12.3	13.2

in any desired spot, generally some refrigerating room. This location offers the advantage that the cooling done by the outside of the cooler, which is of no benefit to the brine traveling through the apparatus, is utilized for cooling the air of the refrigerating room.

Besides the double-pipe cooler, there are a number of other brine coolers on the market, constructed similarly to feed-water heaters, and in which either the brine is contained in the vessel while the ammonia circulates through pipes or tubes with which the vessel is equipped inside, or the ammonia is in the vessel and the brine circulates through the pipes.

The results of several tests made by G. J. Patitz on a double-pipe cooler are shown in the table, page 598. In these tests, the cooler worked under the flooded system.

Atmospheric Brine Coolers.

In localities where the winter is long and the temperature considerably below the freezing point, it is economical to instal an atmospheric brine cooler, i. e., a cooler which is erected where a strong current of air exists, for instance on the roof. The brine running continuously over the cooler is sufficiently lowered in temperature to enable it to do all the necessary cooling.

The cooler is not only very economical, but gives the engineer an opportunity to overhaul the ice machine, which is especially advantageous in small establishments where there is only one machine. A cooler of this kind has the further advantage that its initial cost is very low.

Comparison Between Direct Expansion and Brine Systems.

Both systems have their advantages and disadvantages, and the question which of the two deserves the preference depends on prevailing circumstances and conditions.

With the brine system all the refrigeration can, if desired, be accomplished in the day-time, and consequently only one crew of men is required, while with the direct expansion system, two and in many places even three crews of men are necessary.

The disadvantages of the brine system are the loss by radiation in the brine tank, the greater number of pipes required, the cost of the circulating pump and the power for driving it, and the

lower back pressure which must be carried to produce the same refrigeration.

The fear many people have of ammonia escaping in the direct expansion system with the consequent danger from this source, seems absolutely unwarranted, as but few accidents of this kind are known, and they were mainly due to inferior workmanship and improper installation.

Taking everything into consideration, the brine system may be regarded as more economical for small plants, i. e., for plants of not more than 50 tons capacity, if installed properly, while for plants of larger size, the direct expansion system is to be preferred.

Insulation

The object of insulation is to prevent heat from passing from one body to another. Thus it may serve two distinctly different purposes. In the first place, insulation may be employed for preventing a loss of heat, for instance, through the walls of a steam pipe, and secondly, for avoiding a loss of cold, i. e., a transmission of heat from the outside into a refrigerating room, tank or pipe.

The choice of the proper insulating material is of great importance, and is often a difficult matter. For insulating refrigerating rooms or pipes it must not only be a poor conductor of heat, but must at the same time be air and water-tight.

Should air, and with it moisture, penetrate the insulation, the moisture will condense on the colder parts of the insulating material and on the surface to be protected, and may destroy the insulation and perhaps even the surface, or at least decrease the insulating effect of the protecting cover.

The best insulation is certainly not too good, but sometimes it is a question of economy whether the application of the most efficient insulation, which is usually the most expensive, is advisable. There is, of course, a point where the money invested in a high grade insulation will balance the economy gained in operation. The financial side must, therefore, be carefully considered.

Insulating Materials.

The material used for insulating purposes should be a poor conductor of heat, non-odorous, water-tight, fireproof, non-decaying, easily applicable, and of reasonable cost.

Insulating materials frequently employed in this country are saw-dust, wood shavings, granulated cork, a combination of wood, paper and air spaces, felt lith, impregnated cork, and pure cork-boards, the latter being used extensively. There are many other

substances employed in this country as well as in other countries, such as mineral wool, ground charcoal, kieselguhr, art pumice, cotton, asbestos, flax and hemp preparations, etc. Sometimes several materials are employed in combination, as Professor Ryndall has demonstrated some time ago that heat passes with greater ease through a material of uniform density, than through a mixture or combination of various substances of different densities.

Nearly all of the materials used are of a porous nature, and the fact that they act as insulators depends to a great extent on the presence of the air which is distributed throughout the entire substance and which is an excellent non-conductor of heat.

Cork-boards are used extensively and seem to give the best satisfaction. They are easy applied to walls, ceilings and floors, two layers each two inches thick being generally used for ordinary insulation, and two 3-inch layers for extra ordinary insulation. On brick or stone walls, the first layer is imbedded in soft cement, the latter being first applied to the wall, and the cork-board applied afterward; the second layer is applied to the first layer, using a cement of pitch as a binder between the two layers.

All joints are broken, and on top of the last layer, a cement plaster, in two coats, a rough and a finish coat, is applied, forming thereby a smooth wall. On a wood structure the cork-boards are nailed and then plastered, and on iron tanks the layers are fastened by means of asphaltum pitch and then plastered.

The same method is applied for ceilings and floors, with the exception that for floors a layer of concrete is placed below and on top of the cork.

The heat transmission for pure cork-boards may be taken on the average to be 6.4 B.T.U. per square foot for one inch of thickness, in twenty-four hours, per degree F. difference in temperature between the outer and the inner surface. Taking the heat transmission to be inversely proportional to the thickness, which has been shown to be very nearly correct, the heat transmission for a cork wall 4 inches thick would thus be 1.6 B. T. U. per square foot.

Example.—What is the heat transmission of a brine tank 20 feet long, 14 feet wide and 12 feet high, insulated with 6-inch

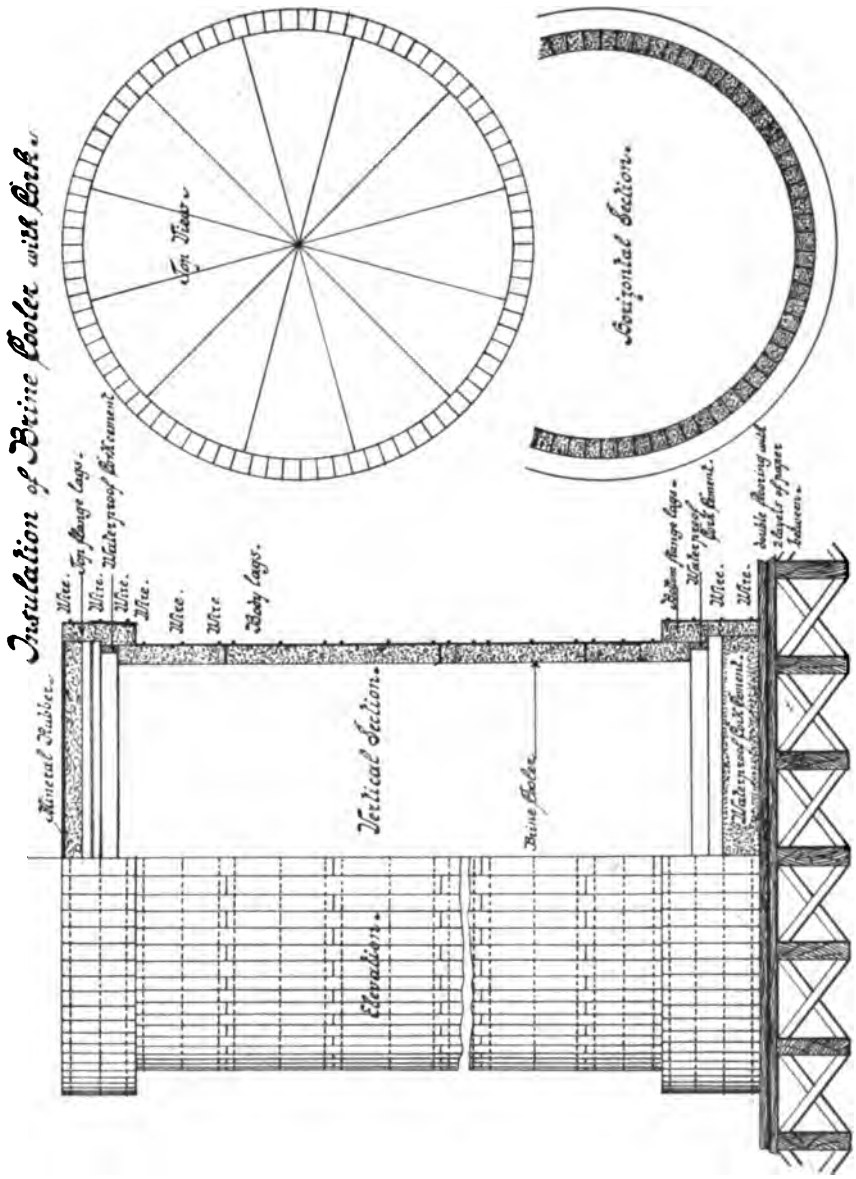


Figure 212. Insulation of Brine Cooler with Cork.

pure cork-board, the brine temperature being 12° F. and the room temperature 92° F.?

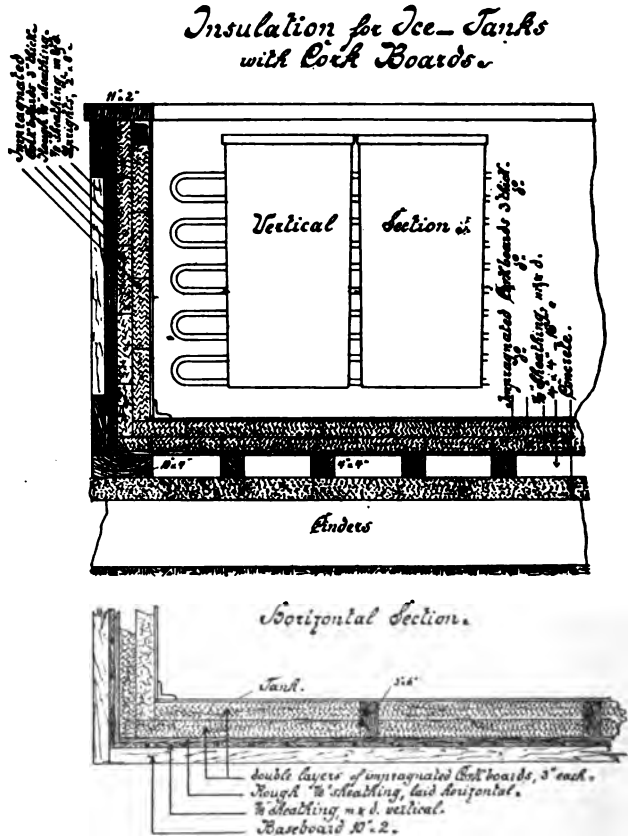


Figure 213. Insulation for Ice-Tanks with Cork-Boards.

Solution.—The tank surface is $2 \times 20 \times 12 + 2 \times 14 \times 12 + 20 \times 14 = 1{,}096$ square feet. $\frac{1{,}096 \times 6.4 \times (92 - 12)}{6} =$ 93,467 B. T. U., or nearly ⅓ ton of refrigeration is lost per 24 hours. If the cork-board were only 4 inches thick, about ½ ton would be lost.

The best non-conductor of heat is a vacuum, next comes confined air, or minute air cells (cork, etc.), the latter being, therefore, generally employed.

The following table, which has been compiled by Peclet, gives the heat transmitting power of various substances, including a number of metals.

Heat Transmission of Various Substances.

In B. T. U. per Square Foot, per Hour, per Inch of Thickness, for One Degree F. Difference between Outside and Inside Temperatures.

Gold	.625	Brick-dust, sifted	1.33
Platinum	.600	Coke, in powder	1.29
Silver	.595	Iron filings	1.26
Copper	.520	Chalk powder, slightly damp	0.90
Iron	.230	Walnut, against grain	0.86
Zinc	.225	Fir, against grain	0.75
Tin	.178	Wood charcoal, in powder	0.64
Lead	.113	Straw, chopped	0.56
Marble, average	24	Coal powder, sifted	0.54
Limestone, fine grained, aver.	14	Mahogany saw dust	0.52
Glass, average	6.6	Wood ashes	0.48
Terra-Cotta	4.8	Canvas, hempen, new	0.42
Brickwork, average	4.8	Calico, new	0.40
Plaster, average	3.8	Writing paper, white	0.35
Quartz sand	2.18	Cotton and sheep's wool	0.32
Oak, against grain	1.7	Eiderdown	0.31
Walnut, with grain	1.4	Blotting paper, gray	0.27
Fir, with grain	1.37	Pure cork board*	0.27
Rubber	1.37		

Efficiency of Insulation.

The efficiency of an insulation depends not only on the nature of the material employed but also and to a considerable extent on the method of application.

The effect of an excellent insulating material may be largely offset by a poor arrangement of the insulation, and on the other hand, if the arrangement is well planned and properly carried out, good results may be obtained with the use of comparatively poor materials. Many insulations have been entirely spoiled by the workmen putting them up. A point which is also often overlooked, and which is of considerable importance, is the proper arrangement of the cooling rooms, inasmuch as those in which the lowest temperature is to be maintained should be placed in the middle. In this manner, the other cooling rooms, which are

*From recent tests.

kept at a somewhat higher temperature, surround and protect the coldest room.

The following table, compiled by Mr. Starr, gives the heat transmission through various insulating structures, and shows the relative merits of the various materials and arrangements.

Transmission of Heat Through Various Insulating Structures.

In B. T. U. per Square Foot, per 24 Hours, for 1 Degree F. Difference in Temperature.

5/8 in. oak, paper, 1 in. lampblack, 7/8 in. pine.....5.7
One 7/8 in. board, 1 in. pitch, one 7/8 in. board.....4.9
Four 7/8 in. spruce boards, two papers, solid, no air space...4.28
Two double boards and paper (four 7/8 in. boards) forming one air space 3.71
One 7/8 in. board, 2 in. pitch, one 7/8 in. board.....4.25
One 7/8 in. board, 2½ in. mineral wool, paper, one 7/8 in. board 3.62
Two 7/8 in. double boards, 2 papers, 1 in. hair felt.....3.318
Two 7/8 in. boards and paper, 1 in. sheet cork, two 7/8 in. boards and paper.....3.3
One 7/8 in. board, paper, 2 in. calcined pumice, paper, and 7/8 in. board.....3.38
Four double 7/8 in. boards with paper between, forming three 8 in. air spaces.....2.7
Hair quilt insulator, four boards, four quilts, hair.....2.51
One 7 in. board, 6 in. pat. silicated straw board, air cell finished inside with thin layer pat. cement.....2.48
One 7/8 in. board, paper, 3 in. sheet cork, paper, one 7/8 in. board 2.1
Two 7/8 in. boards and paper, 8 in. mill shavings and paper, two 7/8 in. boards and paper.....1.35
Same slightly moist.....1.8
Same damp 2.1
Double boards and paper, 1 in. air, 4 in. sheet cork, paper, one 7/8 in. board 1.2
Same with 5 in. sheet cork.....0.9
7/8 in. board, paper, 1 in. mineral wool, paper, 7/8 in. board..4.6
Double boards and paper, 4 in. granulated cork, double boards and paper.....1.7

Steam Pipe Insulation.

Various materials are in use as steam-pipe coverings, the main requirement being that the heat conductivity be poor. The table on page 608 gives the results of several tests compiled by Paulding, and shows the B.T.U. transmitted per hour per square foot per degree F. difference in temperature.

Under average conditions, it may be said that the heat transmission through bare pipe is about 3 B.T.U. per hour per square foot of surface per degree F. difference in temperature. This figure, of course, varies with different sizes of pipe, circulation of air, and other conditions.

The following figures are given by the H. W. Johns-Manville Co. as representing the loss in B.T.U. per *minute* per square foot of exposed surface, under average conditions, and with varying temperature of water and steam:

Steam Pressure Pounds Gauge.	*Temperature Degrees F.*	*Loss per Minute B. T. U.*
Hot Water	175	4.5
Hot Water	200	5.5
5	227	7
10	239	7.5
50	298	10
75	320	11
100	338	12
125	353	13
150	366	13.5
175	377	14
225	397	15
250	406	15.5

Based on this table, they give a method of calculating the saving by covering of steam pipe. Assuming an average coal of a theoretical fuel value of 12,000 B.T.U. per pound, and assuming a boiler and furnace efficiency of 65%, the B.T.U. actually obtained from the coal are 65% of 12,000 = 7,800.

Taking the average steam pressure at 125 pounds, and assuming the plant to run 10 hours a day and 300 days a year, we obtain as the B.T.U. lost per year per square foot of pipe surface: $13 \times 60 \times 10 \times 300 = 2{,}340{,}000$ B.T.U. This is equivalent to

Experiments on Steam-Pipe Coverings.*

Kind of Covering.	Diam. of Test Pipe, Inches.	Thickness of Covering, Inches.	Temperatures F.		B. T. U. per Hour per Square Foot of Pipe Surface.		Date of Test.	Testing Expert.
			Steam.	Air.	Total.	Per Degree Difference.		
Asbestos air cell.....	4	1.12	388.0	72.0	166.0	0.525	1896	Norton
Do	2	0.96	303.3	72.3	165.5	0.716	1901	Jacobus
Asbestos fire felt.....	8	1.30	344.7	79.0	133.5	0.502	1894	Brill
Do	2	1.00	354.7	80.1	198.0	0.721	1896	Paulding
Do	2	0.99	307.4	72.5	180.0	0.766	1901	Jacobus
Asbestos, Navy Brand	2	1.20	309.2	79.4	69.9	0.304	1901	Jacobus
Do	2	1.125	365.2	64.6	176.0	0.585	1901	Barrus
Do	10	1.375	365.2	66.8	112.0	0.375	1901	Barrus
Asbestos sponge felted	2	1.125	364.8	60.7	145.0	0.477	1901	Barrus
Do	10	1.375	364.8	62.8	85.0	0.248	1901	Barrus
Do	2	1.14	309.2	79.4	59.7	0.260	1901	Jacobus
Champion mineral wool	8	1.44	346.1	74.3	86.1	0.317	1894	Brill
Fossil meal	8	0.75	347.1	75.3	238.0	0.876	1894	Brill
Hair felt	2	0.96	302.8	71.4	89.6	0.387	1901	Jacobus
Do	8	0.82	348.3	69.0	117.9	0.422	1894	Brill
Magnesia	4	1.12	388.0	72.0	147.0	0.465	1896	Norton
Do	2	1.09	354.7	80.1	155.8	0.567	1896	Paulding
Do	8	1.25	344.1	66.3	106.6	0.384	1895	Brill
Do	2	1.03	310.9	81.6	69.8	0.304	1901	Jacobus
Do	2	1.00	365.2	64.6	155.0	0.515	1901	Barrus
Do	10	1.19	365.2	66.0	103.0	0.347	1901	Barrus
Manville sectional ...	8	1.70	345.5	78.3	93.4	0.394	1894	Brill
Do	2	1.31	354.7	80.1	157.0	0.572	1896	Paulding
Do	4	1.25	388.0	72.0	143.0	0.453	1896	Norton
Mineral wool	8	1.30	344.1	58.3	81.3	0.284	1894	Brill
Remanit for high pressure	2	1.30	306.6	76.1	83.7	0.363	1901	Jacobus
Remanit for intermediate pressure ..	2	0.88	304.5	73.3	100.3	0.434	1901	Jacobus
Riley cement	8	0.75	347.9	74.3	260.0	0.950	1894	Brill
Rock wool	8	1.60	344.1	63.0	72.0	0.256	1894	Brill

* Paulding "Condensation of Steam in Covered and Bare Pipes."

$\frac{2,340,000}{7,800 \times 2,000} = 0.15$ ton of coal lost per square foot of surface per year. From this figure, when knowing the cost of the coal and the per cent efficiency of the pipe covering, it is an easy matter to calculate the amount of saving obtained by insulating the pipe. The per cent efficiency of various pipe coverings of 1 inch thickness varies from 70 to 86 per cent., the so-called asbestos sponge felted covering and the 85% magnesia covering having an efficiency of from 82 to 86% with 1 inch thickness (see also table, page 608). For greater thicknesses, the efficiency is of course increased, and the efficiency can be readily calculated, when it is considered that the transmission is practically inversely proportional to the thickness. Thus, if a covering of 1 inch thickness has an efficicency of 80%, then the efficiency for 2 inch thickness will be $100 - \frac{100 - 80}{2} = 90\%$.

Ammonia and Brine Pipe Insulation.

Ammonia suction and brine pipes leading though warm rooms are sometimes insulated, in order to avoid a loss of cold and in order to avoid dripping. It has been stated before that the greatest care is necessary when insulating such pipes, inasmuch as the covering must be absolutely impervious to moisture.

Methods of Application.—Various materials are employed, and different methods are followed in their application. One of the methods most generally employed and which is very effective, is to place two layers of heavy, tarred felt tightly around the bare pipe, fasten this properly with wire, cover it with a coat of asphaltum varnish, then put on a layer of one-inch hair felt, fasten with wire, give a heavy coat of asphaltum varnish, put on another layer of one-inch hair felt, then two layers of tarred felt, tie on with wire, cover with a canvas jacket, and finish with two heavy coats of asphaltum varnish. Copper wire is preferable for fastening the material, but twine is sometimes used.

Another method of insulating pipe is to surround it with molds of cork, made in lengths of three feet and split lengthwise. The two halves, after having been applied, are held together by a canvas jacket properly tied on and varnished.

Some firms are using a double layer of sectional covering

with the joints broken and cemented with a water and air-proof compound, the covering consisting of various mixtures of well-known non-conducting substances. Other firms use a thick layer of compressed and molded rock-wool, still others a mixture of hair felt and resin. There is also a covering manufactured which forms two confined air spaces around the pipe, and it is claimed to be very effective.

Insulating Underground Pipes.

Pipes which are layed underground are usually insulated by placing them in narrow boxes made of creosoted planks, the space between the pipe and the wood being filled with a mixture of granulated cork and pitch.

Common Ammonia Compression Machines

There are a large number of different makes of ammonia compression machines manufactured in this country, and a detailed description of all or even most of them would be beyond the scope of this book. For such descriptions, the catalogs of the respective manufacturers must be referred to. The following paragraphs contain only a short description of distinct types of ammonia compression machines commonly employed in the United States.

The Arctic Compressor.

The Arctic horizontal compressor is designated as double-single-acting. The reason for this name is that the gas enters the

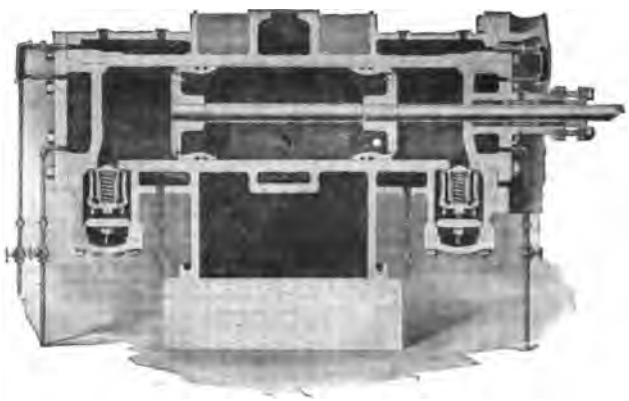


Figure 214. Arctic Compressor.

cylinder at the center, the two pistons on one piston rod each producing an effect equivalent to one cylinder in a vertical single-acting machine.

The suction valve is made of chrome vanadium steel, and has a large area. The discharge valves are located on each end on

the under side of the compressor, and, opening directly downward, allow ready drainage of any entrapped liquid ammonia.

The central portion of the cylinder, where the cold suction gas enters, is cork insulated. The portion of the cylinder representing the last 50 per cent of the stroke, together with the heads, is water-jacketed. The heads are of flat construction, making minimum clearance possible.

The stuffing box is packed in two sections, which are separated by an oil sleeve. This sleeve is supplied with oil by a force feed automatic lubricator, the oil being fed to the sleeve through a ball check valve and a small pipe connection to the under side of the stuffing box. From the top a large pipe provided with a valve connects the stuffing box with the suction side of the system.

The arrangement provides a relief to the oil seal, and serves to feed a small quantity of oil into the compressor. Any ammonia which may leak past the inner set of packing is taken back into the suction pipe. The outer set of packing is subjected to suction pressure only.

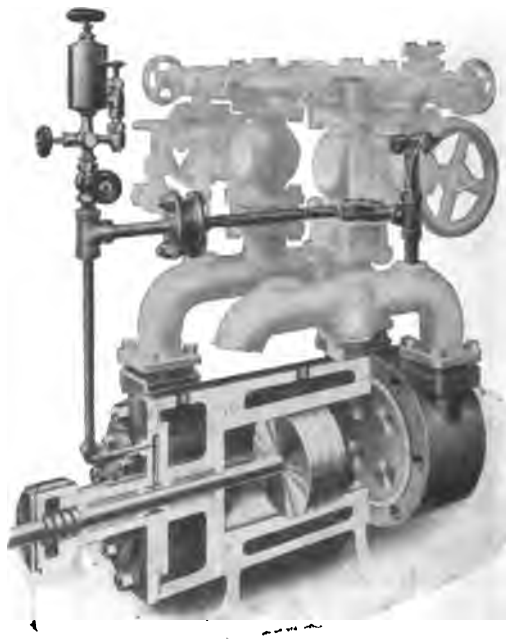


Figure 215. Creamery Package "De Kalb" Compressor.

The Arctic Machine Co. also manufactures a vertical compressor and a carbon dioxide compressor.

Creamery Package Compressor.

The De Kalb compressor is of the horizontal double-acting type. The cylinder heads are flat, and the machine operates with very small clearance. There are four valves in each head, set parallel to the center line. The two suction valves are placed in the two valve ports on the left hand side of the machine, and the two discharge valves are similarly placed on the right hand side. As the discharge valve is on the bottom of the cylinder, any liquid can readily be forced out.

The heads are hollow, forming a water-jacket around the valve chambers and stuffing box, but no water is circulated around that part of the head where the suction gas enters. The cylinder also is water-jacketed, as shown in the figure. The stuffing box is packed in two parts separated by a lantern or oil gland, which provides space for oil seal.

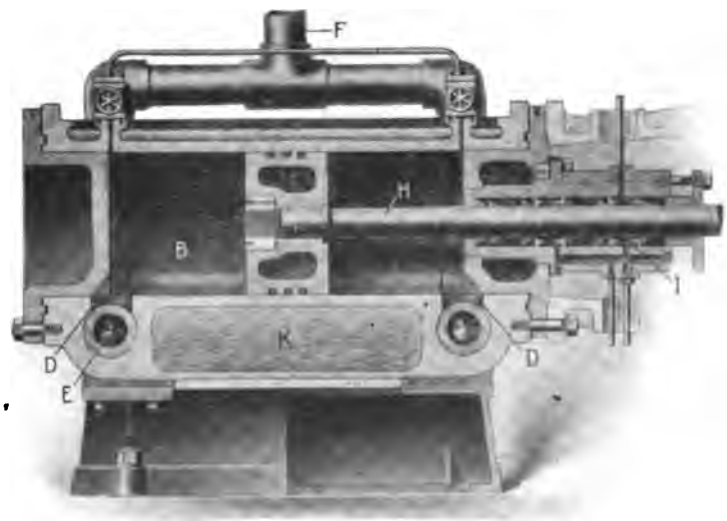


Figure 216. De la Vergne Compressor.

De La Vergne Compressor.

The latest and standard De La Vergne compressor is of the

horizontal double-acting type. The valves in this machine are placed in the bottom of the cylinder, opening in slots at each end.

Referring to Figure 216, the piston rod H enters the cylinder through a gland provided with metallic packing. The packing chamber is provided with an oil gland I, through which is circulated the oil necessary for lubricating the rod. The cylinder is provided with a water-jacket K.

The suction and discharge valves face each other on opposite sides of the recesses D, at a level somewhat lower than the bottom of the cylinder. Thus, no part of the valve mechanism can enter the cylinder. The suction valves when open extend across the recesses D, the width of which is just sufficient to permit proper opening of the valves. The De La Vergne machine is made with a liberal clearance space.

The De La Vergne Machine Co. formerly only made a vertical single-acting compressor, with oil circulation, of which there are many at present in use.

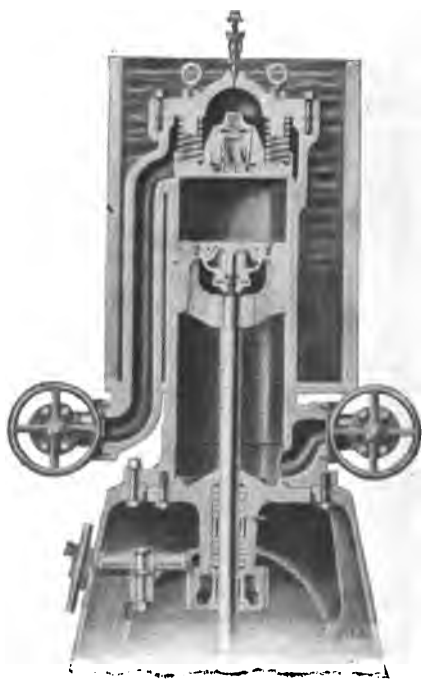


Figure 217. Frick "Eclipse" Compressor.

Frick Compressor.

The Eclipse refrigerating machine built by the Frick Company is of the single-acting vertical type. The cylinders are water-jacketed over practically the entire length of the cylinder and the head. The suction valve is of a large area and is placed in the piston, the gas inlet being at the base of the cylinder. The compressor head is made movable and is therefore called the safety-head. The safety-head does not work as a valve, the real operating discharge valve being the valve in the center. It is self-contained and independent of the cylinder.

The Frick Company also makes a horizontal double-acting compressor.

Huetteman & Cramer Compressor.

The so-called "Safety" machine, manufactured by the Huetteman & Cramer Co., is of the horizontal double-acting type. Its suction and discharge valves are all located below the cylinder, thereby preventing the admission of any broken valve parts into the cylinder. The cylinder heads and piston are flat.

This company also manufactures a vertical double-acting ammonia compressor, and a vertical single-acting compressor.

F. W. Niebling Compressor.

The Niebling Company manufactures a so-called triple suction compressor. This machine is similar in construction to the Linde machine described elsewhere in this chapter, the characteristic difference being that the compressor has three suction and two discharge valves. The third suction valve, called the middle valve, is placed in the middle of the compressor. It is claimed by the manufacturers, that this arrangement allows different back pressures to be carried on the same compressor.

The Vilter Compressor.

The Vilter compressor is of the horizontal double-acting type. The cylinder proper is a semi-steel casting, which is turned and rough bored, then pressed into the cylinder-jacket by hydraulic pressure. The space between the cylinder jacket and the cylinder forms the water jacket. The inside of the compressor

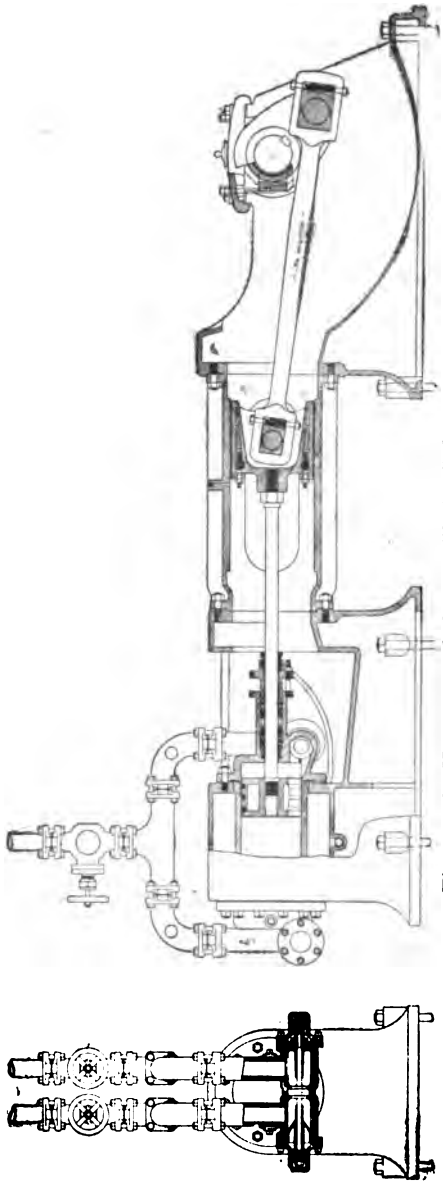


Figure 218. Huetteman & Cramer "Safety" Compressor.

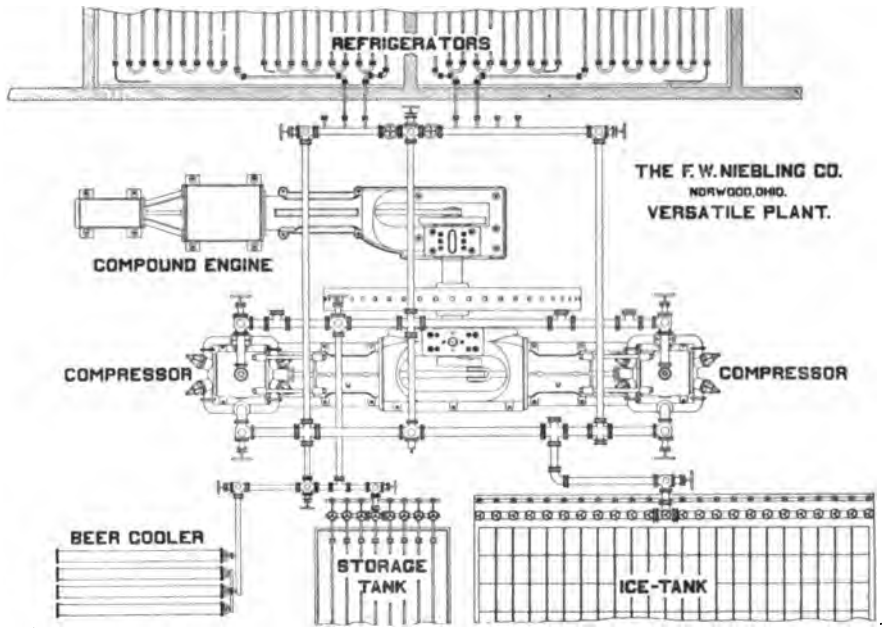


Figure 219. F. W. Niebling "Triple Suction" Compressor.

head is turned to a true semi-sphere, this shape admitting of large valve area. The valve chambers are bored at an angle to the center of the sphere of the heads. These chambers are so fitted

Figure 220. Vilter Compressor.

that when the valves rest in the ports, the seats form a part of the concave surface, reducing clearance between the valve seat and the cylinder to a minimum. In the latest Vilter compres-

sors, the front and back heads of the smaller units are provided with one suction and one discharge valve, of the intermediate units with two suction and two discharge valves, of the largest units with three suction and three discharge valves. The object

Figure 221. Piston, Vilter Compressor.

of the multiple head is to enlarge the total area of valves, which makes it possible to increase the speed, and consequently the capacity of the compressor.

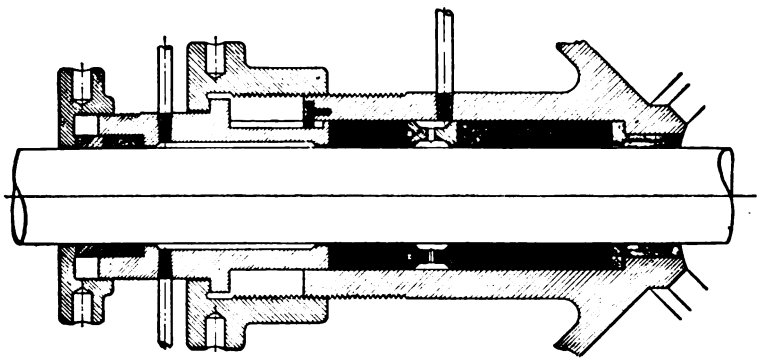


Figure 222. Stuffing Box, Vilter Compressor.

The extremely searching and penetrating character of ammonia gas necessitates particularly tight joints, far superior to those used for steam or air pressures. The Vilter stuffing box, which is shown in Figure 222, is packed with a self-adjusting packing, and an automatic pressure release placed in the first or inner box. This is followed up by a hollow sleeve, through which oil is circulated by an automatic oil pump, thereby lubricat-

ing the piston rod. The outer packing is subjected only to the low pressure of the system. The oil circulated through the sleeve is used over again, only enough being absorbed to lubricate the rod, thus reducing the loss of oil to a minimum.

Wolf-Linde Compressor.

The Wolf-Linde compressor is of the horizontal double-acting type. The piston and heads are spherical and of the same radius, and the space between the head and the piston at the end of the stroke is reduced to 1-32 inch or less. No pockets are formed by the valve ports, and the discharge valves are placed

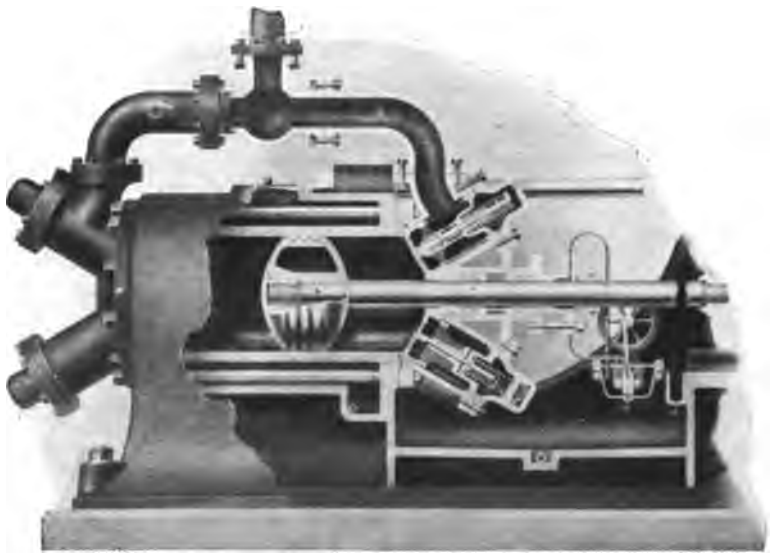


Figure 223. Wolf Linde Compressor.

at the lower point of the cylinder, so that any liquid, which might be drawn into the compressor, will mostly flow out, and is thus prevented from re-expanding before the suction valve can open on the back stroke of the piston. The valve areas are large, and the suction valve is provided with a safety stop, the purpose of which is to prevent its falling into the cylinder.

The stuffiing box consists of two sets of packing, separated by an oil sleeve forming at the same time an oil chamber, where-

by a thorough lubrication of the rod is obtained. However, the main reason for the oil chamber is to prevent the ammonia which may pass by the first set of packing from passing by the second set.

The Linde machine is usually not equipped with a water-jacket, as the ammonia is generally allowed to enter the compressor in a rather wet condition, and the heat of compression is partly taken care of by the unexpanded ammonia, which in the form of a fog entered the compressor on the suction stroke.

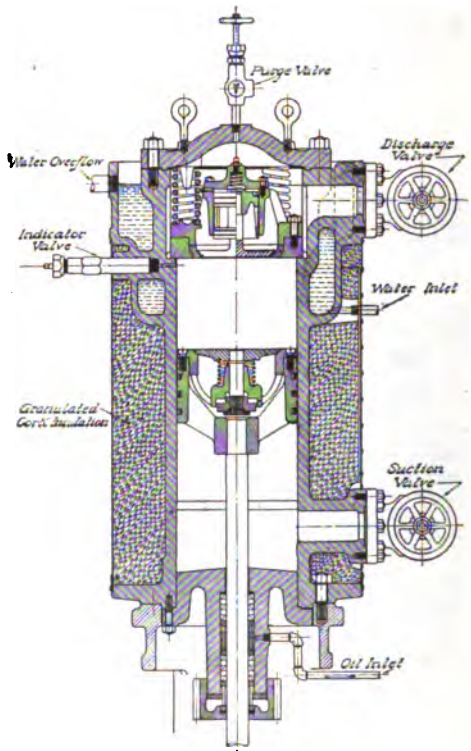


Figure 224. York Compressor.

York Single-Acting Compressor.

This compressor is commonly called the false-head or safety head compressor. The gas enters at the bottom, passes up through the suction valve in the piston, and is compressed and driven out on the upstroke of the piston, through the discharge valve

in the center of the safety head. The piston and the bottom of the safety-head are faced off square and come in contact, thus allowing of a maximum discharge of gas.

It is claimed by the manufacturers that the safety-head secures the compressor against damage, in case of accidental breaking of the valves or any other part of the machine. In such a case, the safety head lifts and allows the obstruction to pass through, or keeps on lifting until it attracts the attenion of the engineer. In case of an overcharge of liquid ammonia, the only result would be hammering of the valves and false heads as they raise to allow the liquid to pass.

The compressor near the discharge end is encased in a water-jacket, the water entering at the bottom, circulating around the walls, and up over the top cover, where it overflows. The water keeps the compressor walls cool, and takes care of a portion of the heat of compression. The cylindrical compressor surface not water-jacketed is insulated with granulated cork, to prevent heat being absorbed by the colder gas entering it.

The York Company also makes a horizontal double-acting machine, and an absorption machine.

Horizontal Double-Acting Compressor. — This compressor is equipped with a water-jacket, the piston and heads are spherical, and turned accurately so as to allow minimum clearance. The suction and discharge valves are large in area. The suction valve is provided with a safety stop to prevent its dropping into the cylinder in case of accidental breakage.

Liquid Air Machines

Old Methods of Liquefaction.

On account of its low critical temperature (—220° F.), air is very difficult to liquefy. Formerly, air was liquefied by subjecting it to high pressure and the action of various cooling agents. Liquefied gases were used successively for cooling, each succeeding one having a lower evaporating point, until the critical temperature was reached, when the air would liquefy.

Siemens was the first to point out the intensifying or regenerative process of liquefying gases having low critical temperatures. He proposed that the air after having been compressed, should first pass through an interchanger and should then be forced to do work in an expansion cylinder. This expansion would chill the air, after which it would again pass through the interchanger, there to meet the succeeding supply of compressed air. In this manner each fresh supply, on its way to the expansion cylinder, would become a little colder than the last one, and the gradual drop in temperature would only be limited by losses due to conduction of heat from outside and the heat produced by friction. Solvay, in 1885, succeeded in reaching a temperature of —139° F. with an apparatus of his invention, but found that at the temperature mentioned, the losses due to conduction of heat, etc., balanced the gains.

Linde Process.

Hampson and Linde were the first to construct the so-called *self-intensifying apparatus*. The former's patents were filed May 23rd, 1895, and Linde's, three weeks later; both men, however, had been working independently. Linde's apparatus was the more practical one, and by its use liquefaction of so-called permanent gases, such as air, oxygen and nitrogen, can be carried out on a large scale and at a moderate cost.

The apparatus is comparatively simple in construction, consisting chiefly of an ordinary compressor, exchanger, water cooler, expansion valve and liquid receiver.

Operation.—The operation is the following: The air is taken in by a water-jacketed compressor, and after compression it is forced through a coil surrounding the compressor and imbedded in the water-jacket, to dispose of the heat of compression. It is then forced through a long coil, the end of which is provided with an expansion valve, and connected to a liquid collecting vessel. From the latter issues another coil which surrounds the one through which the air passed previously, forming a double-pipe coil in which the air is expanded. As the air does interior work when expanding against pressure in the liquid receiver and coil, its temperature is reduced correspondingly.

In this expanded condition the air flows back to the compressor, meeting and cooling on its way, the compressed and consequently hot air coming from the compressor. After the second compression, the air enters the liquid receiver at a lower temperature than before, and by continued expansion and compression it is finally brought to the critical temperature, —220° F., at which it liquefies and remains in the liquid receiver.

The volume in the system, and consequently the pressure, is reduced when the air liquefies, and new air must, therefore, be added by an auxiliary compressor.

To prevent excessive conduction of heat into the cooled air, the double-pipe concentric coils, forming what is called the exchanger, must be well insulated, and they are therefore enclosed in a chamber built of suitable insulating material.

Linde operated his machine between 235 and 2,940 pounds pressure. Since the work required to compress the air depends upon the ratio of the pressures, this ratio should be as small as possible. His machine produces about 4-1/3 pounds of liquid air per horse-power per hour, which represents approximately 5% of the total air handled by the compressor. The first liquid accumulates after the machine has been in operation about two hours.

Principle of Apparatus.—Linde was the first to give numerical data regarding the cycle of operation and principle on which the liquefaction of air is based. He pointed out that the interior work done by the air during expansion is the source of

the refrigeration, which causes the lowering of temperature to or below the critical temperature, at which the air is easily liquefied by pressure. Joule and Thompson demonstrated some forty years ago, that the law of Gay Lussac does not hold good for air and several other gases. The interior work due to expansion was, however, considered very small, and is indeed small at low pressure, while as Linde showed, it increases very rapidly with an increase of pressure; hence the high pressure at which Linde operates his machine. The discovery of Joule and Thompson, that a fall in temperature accompanies a fall from a higher to a lower pressure, is called the *Joule-Thompson effect,* and Linde and Hampson have both used it as the principle on which their machines operate.

The change in temperature which takes place when air is expanded through a fine orifice can be calculated according to the following formula:

$$F = \frac{p - p_1}{4} \times \left(\frac{289}{T}\right)^2$$

F = fall in temperature in degrees Centigrade.
p = pressure in atmospheres (14.7 pounds) before passing expansion valve.
p_1 = pressure in atmospheres after passing expansion valve.
T = absolute temperature of the gas in degrees Centigrade before passing the expansion valve.

From the above formula, it is evident that in order to obtain the largest possible reduction in temperature, the difference between the two pressures p and p_1 must be as great as possible, and the absolute temperature T must be as low as possible. As long as the pressures themselves are low, the difference between them will necessarily also be low, while with an increase in pressures the difference becomes likewise greater.

On the other hand, it does not make any difference with regard to the work required for producing refrigeration, whether the pressures are low or high. As long as the ratio between the pressures remains unchanged, the work required does not vary. The ratio is usually kept at 12.5 to 1, i. e., the pressure p is 12.5 times greater than the pressure p_1, and under these conditions the same amount of work is required whether the air is compressed from 1 to 12.5 or from 16 to 200 atmospheres, but the

refrigeration obtained is much larger when the two pressures are higher.

The influence of the temperature T on the final result can, according to the above formula easily be demonstrated by an example. Supposing for instance, that the starting temperature is 0° C. which corresponds to an absolute temperature of 273° C., and that the pressures of p and p_1 are 200 and 16 atmospheres respectively, then the reduction in temperature will be:

$$F = \frac{200 - 16}{4} \times \left(\frac{289}{273}\right)^2 = 51.55° \text{ C.}$$

If, however, the starting temperature is lower, for instance — 90° C., corresponding to 273 — 90 or 183° C., absolute, then we have:

$$F = \frac{200 - 16}{4} \times \left(\frac{289}{183}\right)^2 = 114.72° \text{ C.}$$

The temperature is reduced considerably lower, or the refrigerating effect is higher.

Commercial Value of Liquid Air.

There is a large amount of energy stored up in liquid air, still for various reasons it is of no practical value for the production of work. The process of liquefying air is very expensive, and even if the cost of production could be reduced, it would have the disadvantage that it cannot be stored with any degree of safety, nor can it be maintained in liquid form for any length of time without losing its greater portion by evaporation.

If merely used for motive power, all of its stored up refrigerating power would be lost.

On account of the enormous cost of production it is entirely out of the question for ordinary refrigerating purposes.

The latent heat of liquid air is only 144 B. T. U. per pound at atmospheric pressure, and the specific heat 0.5. While a pound of ammonia can be deprived of all of its latent heat by one single compression and this at a pressure only 1/18 of that at which air is liquefied, the latter requires a large number of compressions and expansions before a pound of liquid air is obtained.

There are other purposes, however, for which liquid air can be made useful. Linde constructed a special apparatus for the

production of oxygen, based upon the fact that when liquid air is allowed to evaporate, using certain precautions, the nitrogen evaporates first, leaving a liquid which consists of 50% or more of oxygen. This Linde product is very valuable in combustion as well as in various other chemical oxidizing processes, which latter has already been demonstrated in the manufacture of calcium chloride by the Deacon process. Its value for the generation of intense heat during combustion has been proven by its employment for the manufacture of calcium carbide, which previously had only been produced by means of an electric furnace. For various medical purposes it has been successfully applied.

For the generation of a high vacuum in glass vessels it can also be used with very satisfactory results by filling the vessel in which the vacuum is to be created with an easily condensable gas, such as ammonia or carbon dioxide, and then dipping the vessel into liquid air, whereby the gas is liquefied. If the vessel has an extension, this liquid can be collected in it and the extension sealed off, leaving an almost perfect vacuum in the vessel.

Liquid air has also been tried as an explosive, but without very satisfactory results, though recently a successful experiment in this direction is stated to have been made.

Application of Refrigeration

ICE MAKING.

Various methods for the manufacture of ice have been invented, but only a few have proven successful and have been generally introduced. The two common systems are the can system and the plate system, both of which are described in the following pages.

THE CAN SYSTEM.

In the can system of ice making, galvanized iron cans are filled with the water to be frozen, and are placed in a brine tank between rows of pipes through which ammonia circulates. The brine in the tank is kept in motion by a propeller or a pump.

Purification of Water.

If ordinary water were frozen in the cans, the air, which every natural water contains, would give the resulting ice a milky appearance. For the production of transparent ice with a can ice plant, the employment of ordinary water, unless kept in motion during the freezing process, is therefore not practical. Distilled water, practically free from air, should be used, this having at the same time the advantage that the ice made from it contains practically no bacteria and can safely be employed for table use.

For the production of the distilled water, the exhaust steam from the engine and the auxiliary machinery is generally utilized, it being condensed by the water previously used on the ammonia condenser. This condensed water, however, is not sufficiently pure for ice making, containing traces of oil and other impurities, as well as some air, and it must therefore be further purified.

The greater part of the oil should be removed by an ordinary *exhaust oil separator,* before the exhaust steam enters the con-

denser. The latter may be of the submerged, atmospheric or double-pipe type, whichever may appear most practicable under prevailing conditions.

The condensed water coming from the condenser is purified by being passed through a so-called *reboiler,* in which it is vigorously boiled. All of the air is thereby expelled, and the impurities are thrown to the surface, where by the action of the existing current they pass into a *skimming device,* whence they are removed through V-shaped openings and carried into the sewer.

The pure water is discharged at the bottom into the *distilled water cooler,* where its temperature should be lowered as far as possible. The coldest water obtainable should be employed for this purpose. It must be borne in mind that all reduction in temperature after the water has left this cooler must be produced by the ice machine, and that the efficiency of the latter will therefore be the greater the lower the temperature of the water when it leaves the cooler.

From here the water enters the *filter,* either sand or charcoal being used as filtering material. Sometimes an additional sponge or cloth and paper filter is employed with good results. If clear ice is desired, the filters must be cleaned regularly and carefully, and it is therefore advisable to have more than one filter, so that while one is being cleaned the reserve filter can be employed.

After having passed the filters, the water enters the storage tank or *fore-cooler.* This tank is equipped with ammonia coils, whereby the temperature of the water is lowered, and the water then used for filling the ice cans. On account of the formation of ice around the pipes, the efficiency of the ammonia coils is sometimes very low.

In spite of this rather complicated process of water purification, the ice sometimes contains a white core in the center of the block, and sometimes a core of reddish color. Either is a sign of poor filtration. A white core is caused mainly by impurities carried over with the steam from the boiler, and consists principally of calcium and magnesium carbonates; a red core is a sign that corrosion has taken place in the condensing and distilling coils. It is usually noticed after the plant has been standing idle for some time. The best remedy in both cases consists in thorough filtration and violent reboiling.

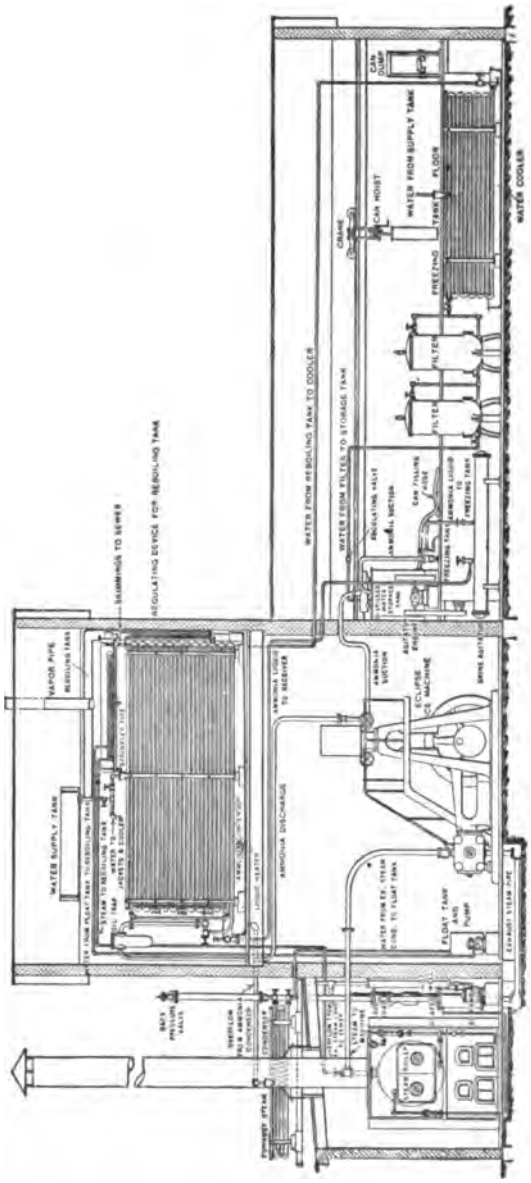


Figure 225. Can System of Ice Making.

Use of Natural Water.

A system was introduced several years ago, according to which clear ice is made from natural water in the can system by simply keeping the water in motion during freezing. This is generally done by forcing air through a small pipe into the water to be frozen. A ¼-inch pipe connected by a small piece of hose to a supply header is inserted into the center of each can. When the ice reaches the end of the pipe, the latter is taken out and replaced by a shorter one, and as the ice keeps growing towards the center, the unfrozen water, when it amounts to about 15 gallons (in a 300-pound can), is drawn out and put in a storage tank. The remaining space is then filled up with a fresh supply of raw water, or in some cases with distilled water. This is done to prevent the formation of a thick white core in each block of ice. If the necessary care is exercised, very good ice can be made from raw water. It can readily be seen that more labor is required for a raw water than for a distilled water can ice plant, but the former has the advantage that no steam is required, so that either electric power or water power can be employed for driving the compressors.

Freezing Tanks.

The best material for the construction of brine freezing tanks is tank steel, and if properly built such tanks will last fifteen years or more.

Wood is preferred by some engineers, and it must be granted that if the tanks are made with the necessary care, their durability is highly satisfactory, as they last practically as long as iron tanks. Wooden tanks are made of 2x4, 4x4, 3x6, or 4x6-inch planks of oak, hard yellow pine, or cedar, the thickness of the planks depending on the size of the tank.

Before being nailed down, the planks should be imbedded in hot asphaltum or pitch, and afterwards lined with ⅞-inch flooring. Where they form the corners, the planks should be dovetailed and the joints filled with pitch or asphalt. In order to prevent the sides from bulging, which would cause leaks, the tanks should be well braced with rods fastened inside. Generally it is not deemed necessary to insulate these tanks, as the wood itself is a poor conductor of heat; it is advisable, however, to cover the outside of the tank with cork board insulation.

Reinforced concrete is also successfully used in the construction of freezing tanks.

The freezing tank should not be larger than is necessary to receive the cans, the refrigerating coils and the agitator. About two inches of space should be left between the cans where there are no pipes, and correspondingly more where the refrigerating pipes pass between the cans, the distance depending of course on the size of the pipes. For the agitator, about 3 feet of additional length is generally allowed for the tank.

Agitator.—The agitator which is employed for keeping the brine in motion in the freezing tank is either an ordinary propeller or a centrifugal pump, generally the former. It is usually driven by a belt, but lately also direct by vertical motors.

Piping of Tank.—The tank should contain twice as much pipe surface for ice making as is required for ordinary refrigerating purposes, so that it ought to be equipped with from 225 to 250 lineal feet of 2-inch pipe or its equivalent in other sizes per ton of ice made. For a flooded system, 175 feet of 2-inch pipe is sufficient.

The pipes should be fed from the bottom. The liquid ammonia generally enters a manifold, and the coils also end with a manifold at the top, from which the ammonia should be drawn into the compressor. For a flooded system, the pipe arrangement is slightly different, as is explained elsewhere (see page 512).

Strength of Brine.—The brine should not be stronger than necessary to prevent its freezing at the lowest temperature required. There is no advantage in making it stronger, as this does not in any way accelerate the freezing process, but merely lowers the freezing point of the brine.

It is believed by many engineers that stronger brine does better work. This belief is based upon the observation they have made that they succeed in lowering the temperature of the brine in a given machine quicker when the brine is strong than when it is weak. The same amount of work, however, has been done in each case, as will appear from the following:

One ton of refrigeration will cool 28,800 pounds of water 10° F., while in the same length of time it will cool an equal number of pounds of calcium chloride brine of a strength of

26%, 12.9° F., because the specific heat of such brine is only 0.778. If this cooled brine then takes up heat again, for instance in the refrigerating room, it is only able to abstract 0.778 B. T. U. per pound for each degree F. its temperature is raised, while water of the same temperature would be able to abstract one B. T. U. per degree F. increase in its temperature. From this it can be seen that the gain by using strong brine is only an apparent one, and that the quicker the temperature of the brine can be lowered by the ice machine, the quicker will its temperature be raised in the refrigerating rooms.

Ordinarily, common salt brine is made 80° salometer strong, and calcium chloride brine 90° to 100° strong.

Ice Cans or Moulds.

The ice cans or moulds should be made throughout of galvanized iron, well riveted and soldered. Cans made of No. 16 gauge material should be turned over top and bottom. Care should be taken that the cans are perfectly straight and not twisted, and that they have a taper such as will permit easy release of the ice.

The larger sizes, i. e., the 200, 300 and 400-pound cans, have ¼x2-inch galvanized bands around the top, while the smaller sizes are provided with ¼x1¼-inch bands; ⅝-inch lifting holes are punched through the bands.

The sizes of cans in common use and in accordance with the standard adopted by the Ice Machine Builders' Association of the United States are given in the following table:

Sizes of Cans.

Weight of Cake of Ice.	Inside Dimensions, Inches.			Length Over All Inches.	Thickness of Material U. S. Standard Gauge.	
	Top.	Bottom.	Length.		Sides.	Bottom.
50 lbs.....	8 x 8	7½x 7½	31	32	No. 16	No. 16
100 lbs.....	8 x16	7¼x15¼	31	32	No. 16	No. 16
200 lbs.....	11½x22½	10½x21½	31	32	No. 16	No. 16
300 lbs.....	11½x22½	10½x21½	44	45	No. 16	No. 16
400 lbs.....	11½x22½	10½x21½	57	58	No. 14	No. 14

For average conditions, from 14 to 16 cans, each of 300 pounds' capacity, should be allowed for every ton of ice produced in 24 hours. This gives the water from 50 to 60 hours to freeze.

The filling of the cans is done by means of an *automatic filler* (Figure 226), which is so constructed as to automatically shut off the water when the can is filled to the desired level. This apparatus fills all the cans to the same height, which may be adjusted to suit any requirement.

Duration of Freezing Process.

In order to obtain the best possible results, the freezing process should be a slow one. The duration depends on the temperature and the circulation of the brine and on the thickness of the ice. The freezing time required for ice of various thicknesses at different freezing temperatures is shown in the following table:

Time in Hours for Freezing Various Thicknesses of Ice at Different Temperatures.

Temperature	Thickness.											
	1 in.	2 in.	3 in.	4 in.	5 in.	6 in.	7 in.	8 in.	9 in.	10 in.	11 in.	12 in.
10° F.....	0.32	1.28	2.86	5.10	8.00	11.5	15.6	20.4	25.8	31.8	38.5	48.5
12° F.....	0.35	1.40	3.15	5.60	8.75	12.6	17.3	22.4	28.4	35.0	42.3	50.4
14° F.....	0.39	1.56	3.50	6.22	9.70	14.0	19.0	25.0	31.5	39.0	47.0	56.0
16° F.....	0.44	1.75	3.94	7.00	11.0	15.8	21.5	28.0	35.5	43.7	53.0	63.0
18° F.....	0.50	2.00	4.50	8.00	12.5	18.0	24.5	32.0	40.5	50.0	60.5	72.0
20° F.....	0.58	2.32	5.25	9.30	14.6	21.0	28.5	37.3	47.2	58.3	70.5	84.0
22° F.....	0.70	2.80	6.30	11.2	17.5	25.2	34.3	44.8	56.7	70.0	84.7	100.0
24° F.....	0.88	3.50	7.86	14.0	21.0	31.5	42.8	56.0	71.0	87.5	106.0	126.0

Harvesting the Ice.

Harvesting of the ice should be done as uniformly as possible; for instance, if 24 tons of ice are withdrawn every 24 hours in 300-pound blocks from the freezing tank, then one block should be drawn every $\frac{300 \times 24 \times 60}{24 \times 2{,}000} = 9$ minutes. This uniformity in drawing the ice is very important, as the brine is thereby kept at an even temperature, cooling of the water is done better, uniform quantities of water are drawn from the storage tank, etc.

After the cans have been lifted from the brine, they are dipped in a tank containing warm water, commonly called the *hot well,* in which the cans are left for a sufficient length of time to permit the ice to loosen. They are then hoisted and placed

in the dumping device (see Figure 227), from which the blocks slide into the storage room.

Some ice manufacturers prefer a sprinkling device to the hot well system. Such a device is illustrated by Figure 228.

A free-moving, light-traveling crane for each tank, with geared hand hoist, makes it easy for the man on watch to take care of 10 to 15 cans per hour. An air hoist capable of handling two cans at a time, or an electric crane capable of handling any desired number of cans, is more frequently used.

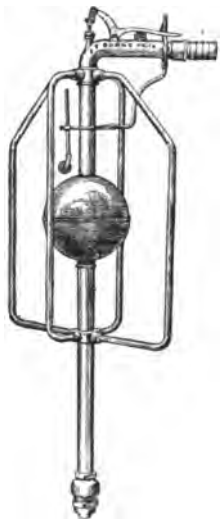


Figure 226. Automatic Can Filler.

Steam Consumption of Can Ice Plant.

Where, as is generally the case, all the water used for can ice making is distilled, the calculation of the steam consumption is very simple. It is only necessary to know the number of pounds of water evaporated in the steam boiler, and then to consider the amount of distilled water wasted, which for medium-sized ice plants amounts to 5 to 10%. The calculation of the steam consumption can be most clearly shown by an example, as follows:

If, for instance, a can ice plant makes 25 tons of ice in 24 hours and wastes 7% of the distilled water, then the total quantity of distilled water required is $\frac{25 \times 2{,}000 \times 100}{93}$ or

approximately 53,750 pounds, and consequently the same number of pounds of steam must be generated by the steam boiler.

Figure 227. Ice Dump.

If the ammonia compressor is driven by a simple Corliss engine, the exhaust steam obtained from the engine together with that of the auxiliary machinery, such as pumps, air compressors, etc., very seldom gives more distilled water than is

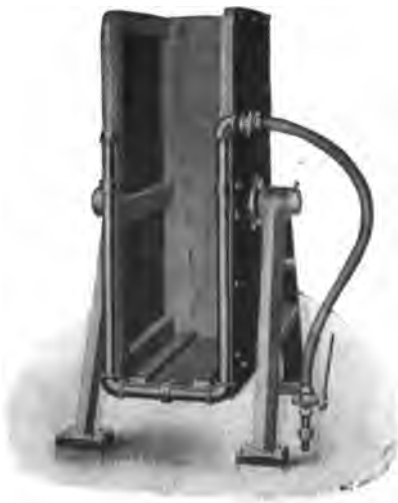


Figure 228. Thawing Apparatus with Sprinkling Device.

required for ice making. It can therefore be said that the quantity of steam furnished by the steam boiler for driving the Corliss engine and auxiliary machinery is approximately equal to the amount of ice produced in 24 hours. In the above case, for a 25-ton ice plant, the steam boiler must therefore evaporate 53,750 pounds of water in 24 hours.

It is now an easy matter to calculate the cost of fuel for producing one ton of ice. If we have determined, for instance, that the fuel for producing 1,000 pounds of steam costs 18 cents, then the total cost of fuel necessary for a 25-ton ice plant will be $\frac{53{,}750 \times 0.18}{1{,}000}$ or $9.62. The cost of fuel for one ton of ice is therefore $\frac{962}{25}$ or 38.5 cents.

In plants of a capacity smaller than 25 tons per day, more steam is generally required to drive the compressor and auxiliary machinery than is needed for the production of the necessary distilled water. In the case of a 15-ton can ice plant, for instance, the author has found the steam consumption to be 35% higher than the quantity of ice produced.

In well-regulated can ice plants, from 6 to 7 pounds of ice are made with one pound of coal, if the latter evaporates 8 pounds of water from and at 212° F. in the steam boiler. It is evident that poor efficiency of the steam boiler must bring about poor efficiency of the ice making plant. Usually too much stress is laid on the performance of the ice machine, and especially of the compressor, and too little on the efficiency of the boiler.

For distilled water ice plants of large capacity, nothing can be gained by installing engines which use the steam at high rates of expansion, since under all circumstances a given amount of water must be evaporated by the boiler to furnish the necessary distilled water. To make ice with a compound condensing engine is, therefore, only economical where sufficient exhaust steam is obtainable from other sources. If the ice plant is connected with a cold storage house, electric power plant, brewery or packing house, such an engine is preferable. Where a compound condensing engine is employed, a surface condenser should be used in connection with it, so that the condensed water can be utilized.

THE PLATE SYSTEM.

In the plate system, hollow steel plates, through which either ammonia or brine circulates in pipes, are immersed in tanks filled with water. Ice gradually forms on both sides of each plate, and since the freezing process is a very slow one, air and im-

purities have an opportunity to escape from the congealing water. For the production of transparent ice, therefore, it is not necessary to employ distilled water, as is generally done in the can system, and natural water can be used with good results.

Purification of Water.

The water used for plate ice making should be odorless, tasteless, clear, and free from pathogenic and putrefactive bacteria. A good drinking water is suitable for the production of plate ice.

If, as is the case in the majority of plants, the water contains appreciable quantities of suspended matter, either organic or inorganic, it must be filtered. Either sand or charcoal serves as filtering material.

Besides freeing the water from suspended matter, filtration serves the purpose of eliminating part of the bacteria present. The micro-organisms which first adhere to the sand grains, if sand is the filtering material, secrete a slimy matter and in this the other bacteria are caught as in a net, so that the water really purifies itself. Still, where large numbers of bacteria are present, it is best to subject the water to a special treatment before passing it through the filter. For this purpose, it is stored in large tanks, and a small quantity of alum, about 0.2%, is added. The water is then stirred either by means of a stirring device or by compressed air, and the precipitate, which forms after the addition of the alum, is allowed to settle. The bacteria are mechanically drawn to the bottom, but the water should be used soon, for if the water is left standing for any length of time the bacteria will rise again.

In order to make a sand filter, a rectangular or round tank of proper size and of suitable material is filled with three layers of gravel and sand. The lowest layer, sharp gravel, is about 18 to 24 inches high, then follows the same thickness of torpedo sand, which is covered by 12 to 18 inches of fine sand. The height of the filter layers is less important than the amount of filtering surface. The latter must be in proper proportion to the quantity of water to be filtered in a given time. If it is too small, the water is not purified sufficiently, since the water must have a greater velocity in order to pass through the filter in a

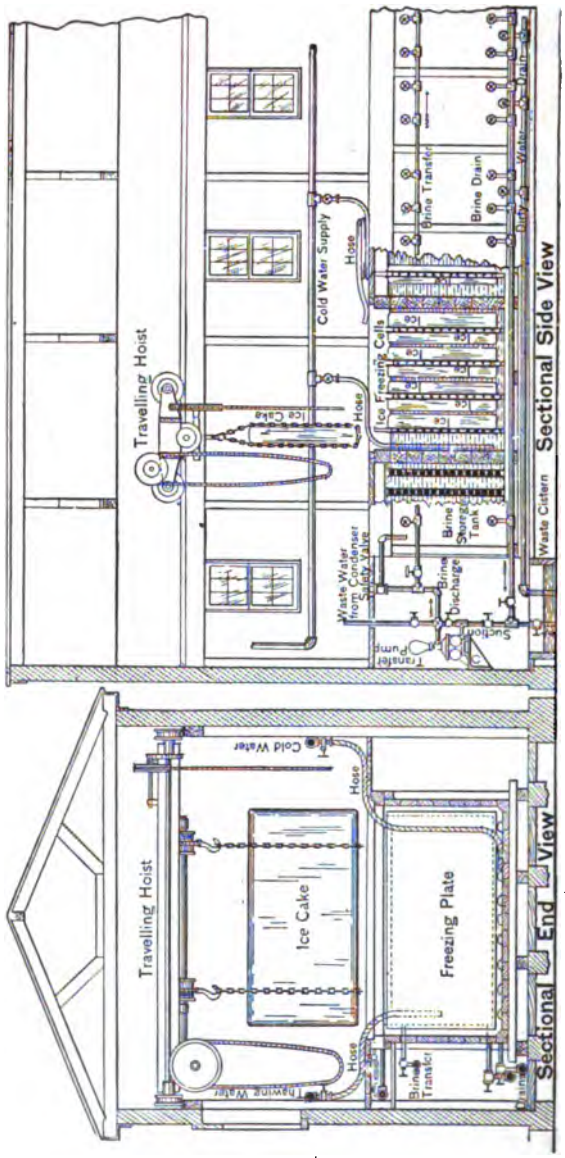


Figure 229. Plate Ice System.

given time, and part of the suspended matter as well as of the bacteria will be carried along. Filtration is most satisfactory when the velocity with which the water travels through the filter bed is 2 inches per hour, but if the water is of medium purity, the velocity can safely be increased to 4 to 6 inches per hour, which corresponds to about 2½ to 3¾ gallons of water per square foot per hour.

From this, if the quantity of water to be filtered is known, the amount of filtering surface can easily be calculated. Let us consider, for instance, a 100-ton plate ice plant which requires 140 cubic feet of water per hour, and assume that the water should be filtered with a velocity of 4 inches per hour. One square foot of filtering surface would then furnish 144x4 or 576 cubic inches of water per hour, and for the filtration of one cubic foot of water (1,728 cubic inches) we should need $\frac{1,728}{576}$ or 3 square feet of filtering surface. Since 140 cubic feet are to be filtered, the total filtering surface must be 140x3 or 420 square feet. This would require a rectangular tank 21x20x5 feet, or a round tub 23 feet in diameter and 5 feet high.

Charcoal filters are generally built about 7 feet high and are filled with charcoal to a height of 5½ feet. Animal charcoal or bone black, and not wood charcoal, should be employed as filtering material. Filtration through charcoal has the advantage over sand filters that it will not only remove suspended impurities, but also any disagreeable taste or odor which the water may possess.

The velocity with which the water runs through these filters may be considerably larger than if a sand filter were employed. Usually about twenty gallons of water are filtered per square foot per hour, which corresponds to a velocity of about three feet per hour.

On account of the rapid filtration, frequent cleaning of the filters is necessary, and it is therefore best to have two or more filters, so that when one of them is being cleaned, filtration need not be stopped.

For purifying the water, filter presses have been used with good success. They consist of perforated plates suitably supported with special filtering cloths between. These filtering

cloths can easily be removed and washed with hot water in a revolving machine. Two sets of cloths are generally used, which assures thorough cleaning and avoids delay.

Freezing Process.

The freezing of the ice in the plate system requires considerably more time than in the can system, partly because all the heat abstracted from the water must be transmitted to the iron plate first and from there to the ammonia coils, and mainly because the formation of ice takes place on one surface only, instead of going on simultaneously on two surfaces opposite each other. The thicker the ice the slower is the transmission of heat through the ice. To obtain a plate of ice of the usual thickness, about 11 inches, 6 to 10 days are required.

The plates are generally 10 to 12 feet high and 14 to 18 feet long, and weigh from 4 to 6 tons, according to thickness.

Harvesting the Ice.

Various methods have been adopted by plate ice manufacturers for thawing the blocks of ice from the iron plate to which they are frozen. Some reverse the current of ammonia so that the hot ammonia instead of entering the condenser enters the refrigerating coils from which the ice is to be loosened. In this case the lower part of the coil must be connected to a liquid receiver, because liquid ammonia rapidly accumulates on account of the heat abstracted from the hot ammonia by the ice block. Other manufacturers circulate hot brine through the hollow plates, while still others employ the warm condenser water.

After the block has been loosened from the plate, it is hoisted by a powerful crane and placed on the cutting table. Here it is cut by circular saws, or grooved by hand plows or by means of a steam cutter, a device consisting of ¼-inch copper pipes through which steam circulates; then it is split into blocks of suitable size and placed in the storage room.

While in the can system ice is drawn throughout the 24 hours, the entire daily output of a plate ice plant is drawn, cut and stored in a few hours.

Kind of Machines to be Used.

Since natural water is used in the manufacture of plate ice, it is most economical and advantageous to drive the compressor

by a compound condensing engine. The auxiliary machinery is best driven electrically; it is then only necessary to drive a dynamo from the engine which operates the ammonia compressor, and this dynamo will furnish power and light for the entire plant. Besides being economical, this method at the same time greatly simplifies the operation of the plant.

Comparison Between Can and Plate Systems.

Since natural water is employed in the manufacture of plate ice, the complicated and costly arrangement for the production of pure, distilled water for can ice manufacture is done away with. Furthermore, a compound condensing engine may be used. On the other hand, the plate system requires more skill for successful operation and costs a little more to be kept in repair. There is from 4 to 5 times as much water exposed to freezing in a plate ice plant as there is in a can ice plant, since the freezing process is a much slower one, and correspondingly more tank capacity is required, which latter fact increases the first cost of a plate ice plant considerably. The initial cost of installation of a plate ice plant is from 25% to 35% higher than the can system, but as the system is considerably cheaper in other respects, as was remarked above, the actual cost of a ton of ice, except in smaller plants, is still 25% less with the plate system than with the can system. It is self-evident that much depends on the size of the plant and on local conditions, and it may be stated that plate ice plants of less than 25 tons capacity are not more economical than a can ice plant of that size.

The following table gives some comparative figures regarding cost of producing one ton of ice with either of the two systems, the can ice plant being taken as 100:

	Can Sys.	Plate Sys.
Hoisting and storing of ice	14.2	2.8
Engineers, firemen and coal passer	15.0	13.9
Coal at $3.50 per gross ton	42.2	20.0
Water pumped directly from a natural source at 5c per 1,000 cubic feet	1.3	2.6
Interest and depreciation at 10%	24.6	32.7
Repairs	2.7	3.4
	100.00	75.4

Season of Ice Manufacture.

In former years, ice plants were shut down during the winter and the production of ice took place during the summer months. Under these circumstances, it frequently happened that during hot weather the demand was greater than the output of the plant, and the question therefore arose as to whether it would be more economical to build a plant large enough to meet the greatest summer demand and to shut the plant during the winter months, or to make the ice during the cold season with a smaller plant and store it. The question seems to have been decided in favor of the latter method.

Ice Storage.

The ice storage rooms are refrigerated and kept at a temperature slightly below the freezing point (26 to 28° F.). As the required refrigeration amounts to from 30 to 50 B. T. U. per cubic foot of contents in 24 hours, the rooms should be piped at a ratio of one lineal foot of 2-inch pipe for every 18 cubic feet of space.

In regard to the best method of storing and packing the ice, there seems to exist a variety of opinions. Some place the ice on edge with every layer crossways, others advise that the ice be stored with alternate ends towards each other and from 1½ to 2 inches apart, the blocks of ice not parallel to each other. In most cases the ice is stored with ½-inch strips between the blocks on top, sides, and bottom, so as to separate them and prevent their freezing together. Between the ice blocks and the walls a space of about 6 inches should be left, which should be well packed with hay or straw. After the house has been filled, the top of the blocks should be covered with either hay, straw, sawdust, soft wood shavings or rice chaff, and of these hay seems to answer the purpose best. In warm climates, however, the ice must be covered at once after coming from the tanks.

The actual storage capacity of an ice house is approximately 50 cubic feet per ton. This allows for the strips between the blocks, their distance from the walls, and for passage ways for the workmen.

COOLING OF ROOMS.

Piping of Rooms.

The piping of refrigerating rooms is frequently a matter of guess work, and in a large number of cases where the refrigerating machine is blamed for unsatisfactory results, the fault lies in reality with the piping of the rooms. Very often the latter are equipped with too small an amount of piping, and poor efficiency and economy of the compressor is the inevitable result.

Pipe Surface Necessary for Given Number of Tons of Refrigeration.

It has been found that one square foot of pipe surface transmits about 2.5 B. T. U. per hour, per degree difference in temperature between the ammonia in the pipes and the air surrounding them. Knowing this, it is an easy matter to calculate the square feet of pipe surface and consequently the pipe length required for a certain number of tons of refrigeration. It is only necessary to know the temperature of the ammonia in the pipe coils, or the suction pressure, and the temperature of the surrounding air. The pipe surface can then be found according to the following formula:

$$S = \frac{T \times 288{,}000}{2.5 \times (t - t_1) \times 24}$$

S = pipe surface in square feet.
T = number of tons of refrigeration in 24 hours.
t = temperature of air in degrees F.
t_1 = temperature of ammonia in degrees F.

Example.—How many square feet of pipe surface are required for a 50-ton refrigerating machine, if the temperature in the cooling rooms is to be 34° F. and if the suction pressure is 25 pounds?

Solution.—From the ammonia table, it is found that the temperature of ammonia corresponding to a suction pressure of 25 pounds is 12° F. The pipe surface must therefore be:

$$S = \frac{50 \times 288{,}000}{2.5 \times (34 - 12) \times 24} = 10{,}909 \text{ square feet.}$$

By multiplying the square feet of pipe surface by the factor 1.6, we obtain the number of running feet of 2-inch pipe necessary. In the above case, we consequently need $10{,}909 \times 1.6$ or 17,454 feet of 2-inch pipe.

Factors which Influence Amount of Piping.

If the suction pressure and consequently the temperature of the ammonia is lowered, the pipe surface may be decreased. For instance, if a suction pressure of 15 pounds, instead of 25 pounds as in the above example, is carried, the corresponding ammonia temperature being 0° F., then the following pipe surface is necessary:

$$S = \frac{50 \times 288{,}000}{2.5 \times (34 - 0) \times 24} = 7{,}059 \text{ square feet.}$$

This corresponds to $7{,}059 \times 1.6$ or 11,294 feet of 2-inch pipe.

In the second case $\frac{(17{,}454 - 11{,}294) \times 100}{17{,}454}$ or 35.3% less pipe surface is required to bring about the same refrigerating effect. At the same time, however, the suction pressure had to be lowered to obtain this reduction in pipe surface, and as stated previously, lowering the suction pressure causes both the capacity and the economy of the refrigerating machine to be reduced.

In a similar manner, the amount of piping is influenced by the temperature of the air in the refrigerating rooms. A glance at the above formula shows that less piping is required when the temperature of the surrounding air is higher and more when it is lower.

Besides the two mentioned above, a great number of other factors play a more or less important role in respect to the pipe surface required for a refrigerating room.

The size of the rooms is of great importance, and the author has found that the required pipe surface closely corresponds to the wall surface of the refrigerating rooms. Since a low and narrow room has a greater wall surface than a high and square room of the same cubic contents, it is evident that more heat will be transmitted in the former case, and more piping will therefore be necessary.

The number of B. T. U. taken up by one cubic foot of space in 24 hours and consequently the amount of refrigeration lost during this time can be seen from the table below. The table has been calculated for rooms of various sizes expressed in cubic feet and for different temperatures inside of the rooms, the outside temperature having been assumed as 80° F., the B. T. U. given

being for each degree difference between outside and inside temperatures.

B. T. U. Lost per Cubic Foot per Degree F. Difference in Temperature in 24 Hours.

Size of Room in Cubic Feet.	Temperature of Room in Degrees F.					
	1°	10°	20°	30°	35°	40°
	B. T. U. lost per cubic foot of space in 24 hours.					
100	2,500	625	470	375	300	235
500	1,600	320	265	200	165	135
1,000	750	150	125	95	80	65
5,000	660	135	110	85	70	55
10,000	570	115	95	70	60	50
25,000	450	90	75	55	46	38
50,000	360	72	60	45	36	30
100,000	250	50	42	32	27	20

Further factors influencing the amount of pipe surface include the following: The minimum inside temperature, the maximum outside temperature, the nature of the goods kept in the refrigerating rooms, the frequency with which these goods are renewed, the more or less frequent opening of doors, the heat given off by the lights used and by the men working in the room, the heat developed by fermentation, for instance in the fermenting cellar of a brewery, the amount of refrigeration necessary to cool the article stored from the temperature at which it is delivered, to the storage temperature, and the heat brought into the room by the water used for cleaning.

All these factors vary to a great extent, and it can readily be perceived that it is a complicated matter to calculate with accuracy the refrigeration required for a certain room and consequently the exact amount of piping. It is, therefore, simpler and better to employ empirical rules, and the statements made in this book concerning the piping of refrigerating rooms are based on empirical rules derived from practical experience.

Tons of Refrigeration Required for Cooling.

If the heat transmitted by walls, floor and ceiling of a room represented the only loss of refrigeration, the calculation of the refrigeration required would be very simple. It would only be

necessary to know the inside and average outside temperature as well as the number of B. T. U. transmitted per square foot of exposed surface in 24 hours.

Example.—A room 80 feet long, 50 feet wide and 20 feet high is to be kept at a temperature of 32° F. The outside temperature is 90° F., and 3 B. T. U. are lost per square foot of wall, floor and ceiling surface per degree F. difference in temperature, this being considered as the only loss. How many tons of refrigeration are required?

Solution.—The exposed surface of the room is:

Floor	$= 80 \times 50 = 4{,}000$ square feet.
Ceiling	$= 80 \times 50 = 4{,}000$ square feet.
Side	$= 80 \times 20 = 1{,}600$ square feet.
Side	$= 80 \times 20 = 1{,}600$ square feet.
End	$= 50 \times 20 = 1{,}000$ square feet.
End	$= 50 \times 20 = 1{,}000$ square feet.
Total surface	$= 13{,}200$ square feet.

The heat transmitted through this surface amounts to $13{,}200 \times 3 \times (90—32)$ or 2,296,800 B. T. U. in 24 hours, and there are consequently necessary for cooling, $\frac{2{,}296{,}800}{288{,}000}$ or about 8.0 tons of refrigeration.

In reality, however, there are a great number of other sources of heat which should be taken into the calculation and which naturally increase the number of tons of refrigeration. Since the contents of the room in the above example are $80 \times 50 \times 20$ or 80,000 cubic feet, one ton of refrigeration cools $\frac{80{,}000}{8.0}$ or 10,000 cubic feet. As a rule, less space is allowed per ton of refrigeration, rarely more. For rooms which should be kept at the freezing point or slightly above, viz., between 32° and 36° F., from 5,000 to 11,000 cubic feet are generally allowed per ton of refrigeration, depending on the size of the room. For breweries of medium size, the average is 8,000 cubic feet per ton of refrigeration.

The number of cubic feet of space that can be refrigerated by one ton in 24 hours, and consequently the number of tons

necessary for cooling a room of given size, can be calculated more satisfactorily according to the following empirical formula:

$$C = \frac{288{,}000 \times c}{s \times k \times (t - t_1)}$$

C = cubic feet of space cooled by one ton in 24 hours.
c = contents of room in cubic feet.
s = surface of walls, floor and ceiling in square feet.
k = B. T. U. transmitted per square foot of surface, per degree F. difference between inside and outside temperature (on the average 3 B. T. U.).
t = maximum outside temperature in degrees F.
t_1 = minimum inside temperature in degrees F.

Example.—A room 40 feet long, 25 wide and 10 high is to be kept at 32° F. The maximum outside temperature is 90° F. and the heat transmitted per square foot of surface per degree F. difference in temperature amounts to 3 B. T. U. How many cubic feet can be cooled by one ton of refrigeration in 24 hours, and how many tons refrigeration are necessary for cooling?

Solution.—The cubic contents of the room are $40 \times 25 \times 10 = 10{,}000$ cubic feet. Its surface is the sum of the following areas:

Ceiling	$= 40 \times 25 =$	1,000 square feet.
Floor	$= 40 \times 25 =$	1,000 square feet.
Wall	$= 40 \times 10 =$	400 square feet.
Wall	$= 40 \times 10 =$	400 square feet.
End	$= 25 \times 10 =$	250 square feet.
End	$= 25 \times 10 =$	250 square feet.
Total surface		= 3,300 square feet.

Therefore, $C = \frac{288{,}000 \times 10{,}000}{3{,}300 \times 3 \times (90 - 32)} = 5{,}016.$

One ton of refrigeration can cool 5,016 cubic feet of space in 24 hours.

Since the contents of the room are 10,000 cubic feet, there are required for keeping the room at 32° F. during 24 hours, $\frac{10{,}000}{5{,}016}$ = approximately 2 tons of refrigeration.

Amount of Piping.

In the majority of plants, the refrigerating pipes are continually covered with a heavy coating of ice, which is caused by the moisture of the air having been, at least partly, deposited on the cold surface of the coils. This coating of ice is detrimental to the cooling effect of the pipes, since ice does not transmit heat so easily as does the bare metal; it acts in fact as an insulating material. If, however, the amount of piping is ample, the ammonia can be turned off for some time in the coils, say for a couple of hours, whereupon the ice will melt. The pipes will afterwards not only perform their duty more thoroughly, but the remaining moisture of the air will also deposit more rapidly on the coils, so that the air will become dry, instead of some of the moisture being deposited on walls and ceiling and causing everything to be damp and often moldy.

In large plants, each refrigerating room can be shut off in this manner at a convenient time and long enough to dispose of the entire ice formed during 24 hours. The best time for doing this is during the night, as this does not inconvenience the men who have to work in the rooms. If the insulation of the room is of average quality, the increase in temperature will be very slight, and when the ammonia is turned on again, the temperature will drop very quickly.

Temporary Stopping of Ice Machine.

In small plants, it is best to stop the ice machine entirely for several hours, preferably at night, when no one is working in the refrigerating rooms and consequently when no refrigeration is lost by such occurrences as opening of doors. The length of time for which the machine can be shut down depends of course on prevailing conditions, for instance, the temperature of the outside air. The author has found, however, that even during the hot season, provided the piping is sufficient, the ice machine can be stopped for from 2 to 6 hours every night. It is an old and mistaken notion that a refrigerating machine must run continuously to produce a given refrigerating effect in cooling rooms, regardless of conditions, and the author has demonstrated in plants where the machine had always been kept running for 24 hours each day that an 18 hours run was sufficient to keep the refriger-

ating rooms at the desired temperature. Supposing the cost of the fuel consumed by the machine per hour to be 60 cents, then $3.60 in fuel expense is saved every day by stopping the machine for six hours.

For an ice manufacturing plant, of course, the conditions are different. The freezing process must be as uniform as possible, and it is therefore not feasible to stop the machine for any length of time.

Number of Inlets and Outlets.

In order to be able to operate the compressor economically, it is essential not only that the pipe surface in the refrigerating rooms be amply large, but also that the pipe coils have no more inlets and outlets than are absolutely necessary. As soon as the number of separate coils, and consequently of inlets and outlets, is too large, the regulation of the flow of ammonia in the system will present numerous difficulties to the operating engineer, and the pipes are either not frosted at all or the frost runs back into the machine. From this it can readily be seen that either the coils or the machine are prevented from performing their work properly. At the same time, the installation of too many separate coils is very costly, since a correspondingly large number of valves and fittings is required.

Proper Length of Coils.

Experience has shown that the most satisfactory results are obtained when the length of 2-inch pipe coils does not exceed 5,000 to 6,000 feet. The proper length, and with it the proper number of inlets or feeds, can be calculated from the diameter of the suction pipe of the machine, the rule being that all the inlets combined should be equal in area to the cross-sectional area of the suction pipe. Consequently, their number can be calculated by dividing the suction pipe cross-sectional area by the cross-sectional area of the pipe used in the coils.

Example.—How many 2-inch pipe coils, and of what length each, must be used for a 50-ton refrigerating machine, if the suction pipe has a diameter of 3 inches? It is assumed that the machine is able to handle the ammonia vapor generated in 15,000 lineal feet of 2-inch pipe.

Solution.—The cross-sectional area of the 3-inch suction pipe

is $3 \times 3 \times 0.7854$ or 7.068 square inches, that of the 2-inch pipe is $2 \times 2 \times 0.7854$ or 3.1416 square inches. By dividing the former by the latter we obtain $\frac{7.068}{3.1416} = 2.5$, so that 3 inlets must be used. The length of each pipe coil must consequently be $15{,}000 \div 3 = 5{,}000$ feet.

For a 50-ton machine there is no necessity of having more than three inlets and three outlets. If we look at the majority of refrigerating plants, however, we shall find not 15,000 feet of 2-inch pipe for a 50-ton machine, but frequently less than 10,000 feet, and not three inlets and outlets, but more than a dozen.

It is needless to say that three inlets can be more easily and better regulated than eight or twelve; still, however natural this looks, it has scarcely ever been carried out in practice, and, as stated before, most of the difficulties experienced can be traced to an improper pipe arrangement.

For this simple reason, a great many refrigerating machines cannot be brought to their full capacity. The motto for the proper installation of refrigerating coils should be: Plenty of pipe surface and long coils. While this rule cannot be strictly followed, it should at least be used as a guide, and no extra feed should be inserted unless it is absolutely necessary.

All fittings should be soldered and all joints flanged. Refrigerating pipes should be properly hung and supported to prevent their shaking and sagging. If long service is expected from pipes in refrigerating rooms, the pipes, as well as hangers, fittings and bolts, should be galvanized. This costs more in the beginning, but is far cheaper in the end, and is also much cleaner. Some engineers have the mistaken idea that ammonia pipes should not be galvanized on the inside, for fear the ammonia would attack the galvanizing and create trouble in the system. However, ammonia does not attack galvanizing any more than it does iron or steel. There are a considerable number of plants which have been equipped entirely with galvanized piping for the last twenty years, and in every case the author was informed of, the pipes gave entire satisfaction and there was not the slightest indication of trouble.

Discs on Refrigerating Pipes.

Some time ago it was thought practicable and economical to increase the refrigerating area of the cooling coils by providing them with round discs. It has been found since that discs are more expensive than if the rooms are supplied with additional piping of equal surface. They have the further disadvantage that they are about four times as heavy as a corresponding amount of piping, require much labor and care when the coils are being cleaned and painted, and to a considerable extent hinder the circulation of air. Taking everything into consideration, they afford no advantage over the plain pipes, and new plants are therefore not equipped with them.

BREWERY REFRIGERATION.

Piping of Cellars.

All the rules given in the preceding pages for the piping of refrigerating rooms in general, apply to the proper piping of brewery cellars also, and only a few special empirical rules will be given here for the calculation of the proper amount of piping of the various cellars and other refrigerated rooms.

Relation Between Floor Space and Refrigeration Required.

A rule which can easily be remembered is to allow 400 B. T. U. for the chip cellar, 700 for the stock cellar, 800 for the fermenting cellar, 900 for the racking room, and 1,600 for the government cellar, per square foot of floor space per 24 hours. This rule applies only to medium-sized breweries, and, as the size of the rooms is not taken into consideration, is only approximate. In small breweries, from 25 to 50% more refrigeration is required. If the rooms are large, high and square, about 25% less refrigeration is sufficient. However, it is always good policy to have about 25% more piping in the refrigerating rooms than the above rule calls for.

Relation Between Piping and Cubic Contents of Rooms.

The following table shows the proper relation between the cubic contents of the cellars and the piping required:

Chip cellar.....1 foot of 2-inch pipe for 22 cubic feet of space.
Stock cellar.....1 foot of 2-inch pipe for 16 cubic feet of space.

Fermenting cellar..1 foot of 2-inch pipe for 17 cubic feet of space.
Settling room .....1 foot of 2-inch pipe for 18 cubic feet of space.
Racking room.....1 foot of 2-inch pipe for 15 cubic feet of space.
Hop storage room..1 foot of 2-inch pipe for 20 cubic feet of space.
Government cellar .1 foot of 2-inch pipe for 8 cubic feet of space.

These figures are correct only in case the direct expansion system is employed. Where brine is used, having a higher temperature than that of the ammonia, the pipe surface calculated according to the above data should be multiplied by 1.25.

If the maximum outside temperature exceeds 90° F., about 10% additional piping is required for each 5° F. increase in temperature. If the outside temperature is lowered, the pipe surface may be decreased in the same ratio.

Lineal Feet of Piping per Ton of Refrigeration.

Another simple method, according to which the amount of piping for a certain number of tons of refrigeration may be calculated approximately, is to allow 600 lineal feet of 2-inch piping per ton of refrigeration. This rule refers to the direct expansion system. For brine cooling, unless the temperature of the brine is the same as that of the ammonia, approximately 750 running feet of 2-inch pipe or its equivalent in other sized pipe is necessary per ton.

Attemperators.

The *fermenting vats* of breweries are generally equipped with attemperators, cooling coils the purpose of which is to keep the temperature of the fermenting wort within certain limits. Even where the brewery is equipped with the direct expansion system, the attemperators are mostly cooled by means of brine, and a small brine tank provided with ammonia cooling coils is, therefore, necessary.

The attemperator coils in the fermenting vat have usually only one convolution and are made of copper or black iron pipe of from 1 inch to 2 inches diameter.

The outside diameter of the coil should be about 12 to 18 inches smaller than the inside diameter of the vat, and the coil hung 18 inches from the top of the tub by three S-shaped hangers. Each coil should be provided with two valves and unions for dis-

connecting in case of emergency. The *pipe surface* required for attemperators in fermenters is about 1 square foot of copper brine pipe surface for every 8 barrels of vat contents. If sweet water, instead of brine, is circulated, about 50 per cent more surface is required.

Attemperators in Chip Casks and Stock Tubs.—In order to control the temperature of the beer in the chip casks and stock tubs, attemperator coils are frequently installed in these vats. The cooling should be done with refrigerated brine as in the fermenters. The coils are usually made of copper. The required pipe surface for chip casks is about one square foot of copper pipe for every 10 barrels of contents, and for stock tubs one square foot for every 12 barrels.

Barrels of Fermenting Wort Cooled per Ton of Refrigeration.

Every pound of sugar changed into alcohol and carbon dioxide in the fermenting vat generates 330 B. T. U., and this heat must be removed by the circulating brine. The total number of barrels of wort cooled by one ton of refrigeration depends, therefore, on the amount of sugar fermented, and can be calculated approximately, according to the empirical formula given below. In this formula, it is assumed that 10% of refrigeration is lost on account of radiation from the brine tank, pump and pipe line. When using this formula, it is not necessary to know the number of pounds of sugar fermented per barrel of wort, but only the percentage of extract the wort contains at the start of fermentation. The formula is as follows:

$$N = \frac{288{,}000 \times 0.90}{E \times 1.5 \times 330}$$

N = barrels of wort cooled per ton of refrigeration.
E = percentage of extract in wort in the cellar.

Example.—How many barrels of wort of 12% Balling (containing 12% of extract) can be cooled in the fermenter by one ton of refrigeration (in 24 hours)?

Solution.—$N = \frac{288{,}000 \times 0.90}{12 \times 1.5 \times 330} = 43.6.$

43.6 barrels can be cooled by one ton of refrigeration.

Tons of Refrigeration Necessary for Given Number of Barrels.

In a similar manner as above the tons of refrigeration necessary to cool a certain number of barrels of wort fermented can be calculated, the formula being:

$$T = \frac{N \times E \times 1.5 \times 330}{288{,}000 \times 0.90}$$

T = tons of refrigeration required.

N = number of barrels of wort fermented.

E = percentage of extract in wort (Balling indication).

Example.—If a brewery ferments 200 barrels of 13% wort per day, how many tons of refrigeration are necessary to cool this amount of wort by means of attemperators?

Solution.— $T = \frac{200 \times 13 \times 1.5 \times 330}{288{,}000 \times 0.90} = 5.$

There are approximately 5 tons of refrigeration necessary.

An empirical rule, which permits calculating the tons of refrigeration for a given number of barrels of fermenting wort with sufficient accuracy, is to allow one ton of refrigeration for 40 barrels of wort in the fermenter. Where a new brewery is being equipped with attemperators, it is advisable to base the calculations on a ratio of one ton per 40 barrels.

Size of Attemperator Brine Tank.

From the number of tons of refrigeration required, the size of the brine tank can be calculated as indicated previously. In the above case, the tank ought to have 5×75 or 375 cubic feet capacity, if used for the attemperators alone. A tank of this capacity is rather small, however, unless the ice machine is kept running almost continually, and it is therefore better to make it larger, so that a considerable amount of refrigeration can be stored in it, and so that it is not necessary to keep the ice machine going all the time. Generally, the attemperator-tanks are made of the same size as the brew kettle, which permits stopping the ice machine for several days. This is especially valuable during cold weather, when no artificial refrigeration is required for the cellars.

Baudelot Cooler.

Baudelot coolers are used for cooling the wort to the temperature desired in the cellar. The cooler consists of a series of horizontal pipes arranged in vertical tiers. The wort to be cooled flows over the outside of the pipes, while through the inside the cooling medium is circulated. The upper rows of pipes, which are of copper, are cooled with water, while the lower pipes are cooled either by means of direct expansion, or by brine. When cooled by ammonia, these pipes are of steel, but when cooled by brine, copper pipe is used. The size of the copper water-cooling pipe is generally 2⅜-inch outside diameter; that of the ammonia cooling pipe, 2-inch inside diameter. Under normal conditions,

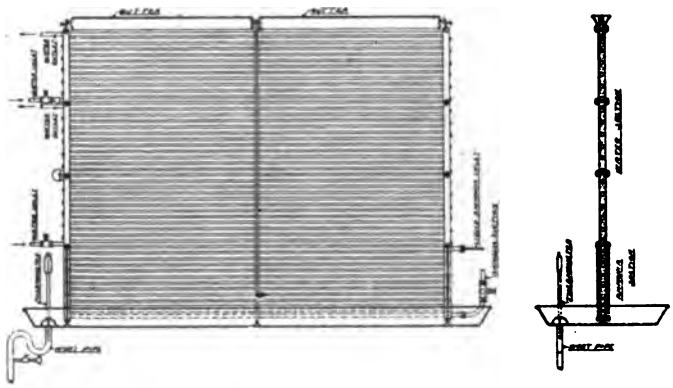


Figure 230. Baudelot Cooler.

a Baudelot cooler should have from 34 to 36 water pipes and from 12 to 16 brine or direct expansion pipes. When proportioned in this way, the cooler should have six square feet of cooling surface per barrel of wort cooled per hour. Figure 230 illustrates the construction of a Baudelot cooler.

The cooling water enters the lowest water pipe, the outlet being at the top. Where the lower part is cooled by means of brine the arrangement is the same, but for direct expansion the ammonia may enter the uppermost ammonia pipe and be drawn off at the bottom, or vice versa. Most of the cooling of the wort must be done by the water, and the ammonia or brine pipes only serve the purpose of removing that part of the heat which the water cannot abstract. Provided the cooler has a sufficient num-

ber of water pipes and the water is forced through in sufficient quantities, the temperature of the wort when leaving the last water pipe should not be more than 2° to 5° F. higher than the temperature of the cooling water.

Capacity of Cooler.—The capacity of a Baudelot cooler is determined by its length and not by its height. For each foot in length of the cooler, from 4 to 5 barrels can be cooled per hour.

Refrigeration Required.—The amount of refrigeration necessary to cool a given number of barrels of wort depends on the temperature with which the wort strikes the first ammonia pipe and the temperature to which it is to be cooled. It can be calculated according to the following formula:

$$R = \frac{n \times 258.5 \times g \times s \times (t_1 - t)}{288{,}000 \times k}$$

R = tons of refrigeration.
n = number of barrels of wort.
g = specific gravity of wort.*
s = specific heat of wort (average 0.91).
t_1 = temperature in degrees F. of wort before striking ammonia section.
t = temperature in degrees F. of wort after cooling.
k = coefficient of performance, generally 85 to 90 per cent.

Example.—How many tons of refrigeration (in 24 hours) are required for cooling 300 barrels of 13% wort from 70° F. to 45° F.? The coefficient of performance is 88%.

Solution.—The specific gravity of a wort of 13% Balling is 1.053. The following number of tons of refrigeration in 24 hours are required:

$$R = \frac{300 \times 258.5 \times 1.053 \times 0.91 \times (70 - 45)}{288{,}000 \times 0.88} = 7.34 \text{ tons.}$$

A reliable and simple empirical method for calculating the tons of refrigeration required for cooling a certain number of barrels, is to allow one ton of refrigeration for cooling 100 barrels 10° F. If this rule is applied to the above example we have:

$$R = \frac{300 \times (70 - 45)}{100 \times 10} = 7.5 \text{ tons}$$

It will be perceived that this tallies very closely with the result obtained according to the first method.

* For specific gravity of worts of different Balling indications, see table, page 45.

Capacity of Ice Machine for Cooling Wort in Given Time.—The result obtained above is the required capacity of the machine if cooling were to be done in 24 hours. In practice, however, cooling must be done in a shorter time, and hence a correspondingly larger machine is necessary. For instance, if in the above example, cooling of the 300 barrels of wort should be done in 3 hours, the capacity of the machine would have to be:

$$\frac{7.5 \times 24}{3} = 60 \text{ tons.}$$

If, however, the brewery has only a 50-ton machine at its disposal, it is self-evident that cooling of the 300 barrels of wort will require more time, viz., $\frac{7.5 \times 24}{50}$ or 3.6 hours.

Influence of Temperature of Wort on Capacity of Machine.—It has been remarked before that the amount of refrigeration required depends among other factors on the temperature with which the wort strikes the first ammonia pipe. It is self-evident that the capacity of the machine is affected accordingly. For instance, if the wort should strike the ammonia pipes at 95° F. instead of 70° F. as in the above example, the required capacity of the ice machine for cooling in 3 hours would have to be:

$$\frac{300 \times (95 - 45) \times 24}{100 \times 10 \times 3} = 120 \text{ tons.}$$

The capacity of the ice machine must be twice as large as under the conditions of the first example.

Double-Pipe Coolers.

Instead of a Baudelot cooler, a so-called double-pipe or enclosed cooler is frequently employed for cooling wort. Such a device is illustrated by Figure 231. This type of cooler consists of a series of tubes through which the wort flows, and another similar series of larger tubes enveloping the smaller ones, thus forming an annular sleeve, through which the cooling media are passed.

The water section of this cooler is usually made of 3½-inch copper wort tubes tinned on the inside, which are enveloped by 4½-inch copper water tubes. The ammonia section consists of 3-inch steel wort pipes, tinned on the inside, and enveloped by 4-inch steel ammonia pipes.

The cooling surface should be 8 square feet per barrel of wort cooled per hour, of which two-thirds should be water cooling surface and one-third ammonia cooling surface.

Tests on Double-Pipe Wort and Beer Coolers.—On pages 660 to 662 are given the results of thorough tests on a double-pipe wort cooler, an ammonia beer cooler and a brine beer cooler, made by G. J. Patitz. The tests extended over a long period of time, and the results obtained are therefore very valuable. The mean temperature differences are calculated in accordance with the Grashof formula as quoted by Hausbrand as follows:

$$T_1 = \frac{T_2 - T_3}{\log_n \frac{T_2}{T_3}} = \frac{(t_1 - t_4) - (t_2 - t_3)}{\log_n \frac{t_1 - t_4}{t_2 - t_3}}$$

T_1 = mean temperature difference.
T_2 = difference in temperature of the two liquids at the inlet end.
T_3 = difference in temperature of the two liquids at the outlet end.
t_1 = inlet temperature of the liquid to be cooled.
t_2 = outlet temperature of the liquid to be cooled.
t_3 = inlet temperature of the cooling liquid.
t_4 = outlet temperature of the cooling liquid.
$\log_n$ = hyperbolic logarithm.

$$T_2 = t_1 - t_4$$
$$T_3 = t_2 - t_3$$

Calculating the mean temperature difference of the test in first column of the water-section test, we have:

$t_1 = 183°$ F. $\qquad t_3 = 34°$ F.
$t_2 = 68°$ F. $\qquad t_4 = 120°$ F.
$T_2 = t_1 - t_4 = 183 - 120 = 63°$ F.
$T_3 = t_2 - t_3 = 68 - 34 = 34°$ F.

$$T_1 = \frac{63 - 34}{\log_n \frac{63}{34}} = \frac{29}{4.1431 - 3.5264} = \frac{29}{0.6167} = 47° \text{ F.}$$

Instead of subtracting the logarithms, we may also obtain the result by dividing 63 by 34 and then taking the logarithm of the quotient. Thus, $\frac{63}{34} = 1.8529$. Hyperbolic log. $1.8529 = 0.6167$.

The arithmetical mean temperature difference in this instance would be:

$$\frac{63+34}{2}=\frac{97}{2}=48.5^\circ \text{ F.}$$

A comparison of the mean temperature differences (logarithmic) and the heat transmission per square foot, as determined in these tests, as well as tests on a brine cooler and a brine tank (see pages 596 and 598), follows:

	Mean Temperature Difference	B. T. U. per Sq. Ft. per Hour per 1° F. Difference in Temperature
Water section, wort cooler	43.4° F	103
Ammonia section, wort cooler	33.5° F.	57.7
Ammonia cellar beer cooler	33.9° F.	20.8
Brine cellar beer cooler	9.6° F.	76.5
Brine cooler, flooded system	13.7° F.	89.7
Brine tank, submerged coils	9.8° F.	30.7

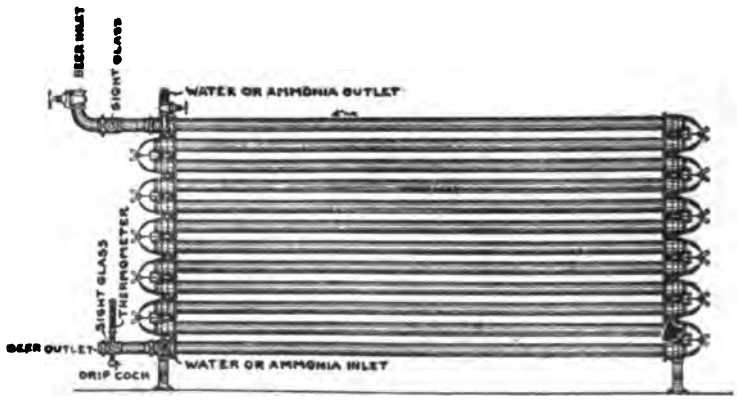


Figure 231. Double-Pipe Cooler.

In these tests, it is necessary to state that the cooler pipes of the wort coolers and the ammonia beer cooler were seamless drawn steel tubes. In the brine beer cooler, the inner tubes were of copper. Tests 1, 2, 3 and 4 on the ammonia beer cooler were made when the cooler was used as an auxiliary to the brine beer cooler, and therefore do not show the best possible results. Tests 5 and 6, however, were made under forced conditions, to ascertain the utmost capacity.

Tests on Double-Pipe Hot Wort Cooler.* Water Section (6 Coils).

	Dec.	Jan.	July	Oct.	Jan.	July	Jan.	Aug.	Dec.	Average
Initial temperature of hot wort, degrees F.	183	181	185	180	185	180	164	182	174	179
Final temperature of hot wort, degrees F.	68	66	79	73	64	85	57	84.5	70	72
Wort cooled, degrees F.	115	115	106	107	121	95	107	97.5	104	107
Initial temperature of cooling water, degrees F.	34	33	63	57	33	67	34	70	46	49
Final temperature of cooling water, degrees F.	120	113	111	106	106	108	89	113	83	109
Water warmed, degrees F.	86	80	48	49	73	41	55	43	37	60
Ratio of wort to cooling water, wort = 1	1.2	1.4	2.4	2.15	1.9	2.32	1.94	2.27	2.8	1.78
Difference in temperature between initial wort and final water	63	68	74	74	79	72	75	69	91	70
Difference in temperature between initial water and final wort	34	33	16	16	31	18	23	14.5	24	23
Barrels of wort cooled per hour	220	220	157	156	172	145	184	176	140	175
B. T. U. per hour in water section	6,325,000	6,325,000	4,160,500	4,160,500	5,203,000	3,443,750	4,922,000	4,290,000	3,640,000	4,719,750
B. T. U. per hour per square foot in water section	6,000	6,000	3,940	3,940	4,925	3,261	4,661	4,060	3,447	4,470
B. T. U. per hour per square foot per 1° F. difference in temperature	128	124	104	104	96	84	106	116	69	103
Mean temperature difference, degrees F.	47.0	48.4	37.9	37.9	51.3	39.0	44	34.9	50.3	43.4
Velocity of wort in feet per minute	132	132	62	62	71	58	74	70	56	80
Velocity of water in feet per minute	135	135	128	114	112	113	121	136	133	125
Combined velocity per minute	267	267	190	176	183	171	195	206	189	205
Cooling surface in square feet	1,056	1,056	1,056	1,056	1,056	1,056	1,056	1,056	1,056	1,056
Tons of refrigeration per 24 hours	535	535	350	350	432	293	415	375	312	400

* Made by G. J. Patitz.

Ammonia Section (4 Coils).

	Dec.	Jan.	July	Oct.	Jan.	July	Jan.	Aug.	Dec.	Av.
Initial temperature of wort, degrees F.	68	66	79	73	64	85	57	84.5	70	72
Final temperature of wort, degrees F.	40	40	40	40	40	40	40	46.5	40	41
Wort cooled, degrees F.	28	26	39	33	24	45	17	38	30	31
Refrigeration in tons per 24 hours	132	130	129	129	84	125	66	146	90	115
Square feet of cooling surface	704	704	704	704	704	704	704	704	704	704
Temperature of ammonia, degrees F.										
Mean temperature difference, degrees	...	...	...	...	...	...	...	...	...	21
F.	...	...	...	...	...	...	...	...	...	33.5
B. T. U. per hour per square foot	...	...	...	...	...	...	...	...	...	1,933
B. T. U. per hour per square foot per 1° F. difference in temperature	...	...	...	...	...	...	...	...	...	57.7

Tests on Brine Beer Cooler (2 Coils).

Test Number	1	2	3	4	5	6	Average
Barrels of beer cooled per hour	165	178	177	166.5	207	134	171.2
Initial temperature of beer, degrees F.	40.1	39.4	40.8	40.6	40.6	37.1	39.7
Final temperature of beer, degrees F.	31.9	32.2	33.3	32.6	32.8	32.5	32.5
Cooled down, degrees F.	8.2	7.2	7.5	8	7.8	4.6	7.2
Initial temperature of brine, degrees F.	26	22.5	18.5	19.6	20.6	19	21.03
Final temperature of brine, degrees F.	33.7	31.6	31.6	31.5	31.2	30.5	31.6
Brine warmed up	7.7	9.1	13.1	11.9	10.6	11.5	10.65
Difference beer inlet, brine outlet, degrees F.	6.4	7.8	9.2	9.1	9.4	6.6	8.08
Difference beer outlet, brine inlet, degrees F.	5.9	9.7	14.8	13.0	12.2	13.5	11.5
Square feet surface	440	440	440	440	440	440	440
B. T. U. per square foot per hour	787	745	772	820	940	358	737
Mean temperature difference, degrees F.	6.1	8.7	11.8	10.9	10.7	9.6	9.63
B. T. U. per degree F. per square foot per hour	129	85.6	65.4	75.2	87.9	37.3	76.5
Refrigeration in tons per 24 hours	29.28	27.6	28.8	31.2	34.8	13.2	27.48
Velocity of beer, feet per minute	198	213	212	200	284	161	211.33
Velocity of brine, feet per minute	350	293	250	293	293	203	280.33
Combined velocity, feet per minute	548	506	462	493	577	364	492

Tests on Ammonia Cellar Beer Cooler (2 Coils).

Test Number	1	2	3	4	5	6	Average*
Initial temperature of beer, degrees F.	32.7	34.5	38	34	39.2	39.2	39.2
Final temperature of beer, degrees F.	31.2	31.6	33	31.6	35.6	33.8	34.7
Ammonia pressure on cooler, pounds per sq. in.....	27.9	28.2	29	28.7	17.5	19	18.2
Ammonia gas temperature, degrees F.	+15	+15	+16	+16	+2	+4	3.0
Beer cooled down, degrees F.	1.5	2.9	5	2.4	3.6	5.4	4.5
Square feet surface.....	300	300	300	300	300	300	300
B. T. U. per square foot per hour	211	410	755	424	660	752	706
Mean temperature difference, degrees F.	17	18	19.4	16.8	35.4	32.4	33.9
B. T. U. per 1° F. difference per hour per square foot..	12.4	25.5	38	23.8	18.6	23.1	20.8
Refrigeration in tons per 24 hours	5.28	11.04	19.20	10.56	19.2	20.04	19.62
Velocity of beer, feet per minute	198	213	212	284	237	182	209
Barrels of beer per hour....	165	178	177	207	216	167	191

Malt-House.

In order to enable the maltster to make malt during the whole year without stopping the plant during the hot season, refrigeration has been introduced. It is essential that the air in the germinating rooms be kept at a sufficiently low temperature, and this can be achieved, even if the outside temperature is very high, by cooling the air in refrigerating chambers.

This cooling is usually accomplished by means of water. In case the temperature of the water is too high for this purpose, the water may be cooled by means of a refrigerating machine.

COLD STORAGE.

Refrigeration of cold storage houses consists in keeping the refrigerating rooms at the required temperature.

Rules for Piping.

The general rules for piping given previously also hold good for cold storage.

When calculating the necessary amount of refrigeration and of piping, it must be taken into consideration that the heat to be abstracted from the storage rooms consists of the heat which

* Figured from tests 5 and 6.

enters through walls, floor and ceiling, the heat given off by the goods to be stored and the heat contained in the air used for ventilating. A simple empirical rule is to allow the following amount of piping:

Storage rooms—One foot of 2-inch pipe for 25 cubic feet of space.

Freezing rooms—One foot of 2-inch pipe for 7 cubic feet of space.

Fish freezing rooms—One foot of 2-inch pipe for 2 cubic feet of space.

These figures are correct for the direct expansion system. Where brine is used, correspondingly more piping must be installed, unless the temperature of the brine is the same as that of the ammonia, in which case the above amounts of piping are sufficient.

Ventilation.

The air in the refrigerating rooms must be renewed from two to four times a day, depending on the nature of the goods. Thorough ventilation is especially necessary in the chilling and storing rooms for meat, and such goods which are liable to generate malodorous gases.

In some cases, where the goods are frequently brought in and taken out, a sufficient change of air is obtained by the opening of the doors, if necessary assisted by opening of the windows. Some cold storage houses have their cooling rooms equipped with ventilating shafts which are generally located in the ceiling, other plants produce thorough ventilation by means of a fan system, while still others have small electric fans directly in the storage rooms.

Proper Humidity of Air.

An important factor in the storage of various goods is the maintenance of the proper humidity of the air. If the air is too moist, formation of mold is liable to result; if too dry, the goods will shrink. For the latter reason it is not advisable to employ cold air refrigeration in cold storage houses, though it is sometimes employed in packing houses.

The humidity of the air should if possible vary with the storage temperature; the lower the latter, the greater should be

the amount of moisture the air contains. For the relation of humidity and moisture in the cold storage of eggs, where it is especially important, see that heading.

Disinfection.

Damp air in conjunction with a comparatively high temperature is liable to cause a growth of mold in the storage rooms. This can be avoided, however, if the rooms, besides being thoroughly ventilated, are white-washed at regular intervals. In some cases, for instance in the storage of meat, it may become necessary to use stronger means of disinfection. Good results are obtained by the use of formaldehyde vapors, which are generated in the rooms by a specially constructed lamp in which wood alcohol is burned.

Storage Temperatures.

The various articles require certain temperatures in order to be kept in the best possible condition, and a temperature that is low enough for one article may be entirely too high for another. The average temperature of cold storage rooms is approximately 34° F., while the temperature of freezing rooms varies from 0° to 10° F., and in some cases is as low as 20° F. below zero. Many experiments have been made in order to determine the best storage temperaure for each individual article, but the results arrived at by the different investigators vary to a greater or less extent. The table on page 665 gives the extremes in storage temperatures of various goods, as given by different investigators.

Separate Storage of Various Goods.

Most articles require separate rooms for storing, either on account of the difference in temperature at which the various goods must be kept or because of their sensitiveness to foreign odors. This is another reason why cold air cooling is not suited to cold storage of a variety of articles, unless the cold air for the different rooms is produced in separate refrigerating chambers.

In the following paragraphs will be given a short description of the methods of storing the most important articles.

Eggs.—The storing of eggs is generally done during the month of April or the early part of May. The fresher the eggs

Cold Storage Temperatures for Various Goods.

Ale	33—42
Apples	32—35
Apple and Peach Butter	40—42
Asparagus	33—35
Bananas	34—45
Beans (dried)	32—40
Beef (fresh)	35—39
Beef (dried)	36—45
Beer (in casks or barrels)	33—42
Beer (in bottles)	45
Berries (fresh)	36—40
Buckwheat flour	40—42
Butter	14—38
Butterine	20—35
Cabbages	32—35
Cantaloupes	40
Carrots	33—35
Celery	32—35
Cheese	28—35
Chestnuts	33—40
Chocolate (to cool)	40
Cider	30—40
Cigars	35—42
Clarets	45—50
Corn Meal	42
Corn (dried)	35—45
Cranberries	33—36
Cream	35
Cucumbers	38—40
Currants	32
Dates	45—55
Eggs	29—35
Ferns	28
Figs	55
Fish (fresh)	20—30
Fish (fresh water, frozen)	17—20
Fish (salt water, frozen)	14
Fish (canned)	35
Fish (dried)	35—40
Fish (to freeze)	5
Flour	36—40
Fruits (dried)	35—40
Fruits (canned)	30—35
Furs (undressed)	35
Furs (dressed)	25—35
Game (frozen)	25—28
Game (to freeze)	0—28
Game (long storage)	10
Ginger ale	36
Grapes	32—40
Hams	20—35
Hogs	30—33
Honey	36—45
Hops	32—35
Lard	34—45
Lemons	33—45
Liver	20—30
Maple syrup sugar	40—45
Margarine	18—35
Meat (brined)	35—40
Meat (canned)	30—35
Meat (fresh)	34—40
Melons (for 3 or 4 weeks)	35
Milk	32
Mutton (fresh)	33—36
Mutton (frozen)	25—28
Nuts (in shells)	35—40
Oatmeal	40—42
Oil	35—45
Oleomargarine	20—35
Onions	32—40
Oranges	34—50
Oysters	33—35
Oysters (in tubs)	25—35
Oysters (in shells)	30—43
Ox tails	32
Parsnips	32—35
Peaches	34—55
Pears	40—45
Peas (dried)	32—40
Plums	33—42
Pork	34—40
Porter	34
Potatoes	20—30
Poultry (frozen)	5—22
Poultry (to freeze)	10
Poultry (long storage)	35—40
Sardines (canned)	35—38
Sauerkraut	20—38
Sausage casings	40—45
Sugar, etc.	35
Sweet corn	35—45
Syrup	30—35
Tenderloin, butts, ribs	34—42
Tobacco	32—36
Tomatoes	35—42
Veal	34—40
Vegetables	34—40
Watermelons	40—42
Wheat flour	40—50
Wines	25—45

the better it naturally is, and before putting them into the cold storage rooms, the eggs are therefore tested as to their age. In the storage of eggs, the humidity of the air, and consequently the ventilation, are of importance. Madison Cooper gives the following relation of temperature and relative humidity:

Temperature Degrees F.	Relative Humidity Per Cent.	Temperature Degrees F.	Relative Humidity Per Cent.
28	80	35	65
29	78	36	62
30	76	37	60
31	74	38	58
32	71	39	56
33	69	40	53
34	67		

For storing a three-dozen case of eggs, about 3 cubic feet of space are required, which allows for passage ways and space between the goods and the ceiling. The average storage temperature is 33° F.

Fish.—Before being put into the storage rooms, fish must be frozen. This is usually done in galvanized iron pans 22 inches long, 8 inches wide and 2½ inches deep, covered with lids, each pan containing about 12 pounds of fish. The pans are put into the so-called *sharp freezers* and kept there for from 72 to 84 hours, exposed to a temperature ranging from 10 to 16° F. below zero.

Freezing could be done in a shorter time, but not economically, as lower temperatures and consequently a lower back pressure would be required. The sharp freezers are vaults provided with iron doors and contain the cooling coils arranged in the form of shelves on which the freezing pans are placed. The capacity of the freezing chambers is generally about 2½ tons.

Some fish which are very fat generally show small oil spots on their surface after freezing, which are caused by the rapid contraction of the fish meat during the freezing process. Through long storage or careless handling of the fish, these small oil drops may become rancid. In order to prevent this, the fish, when com-

ing from the sharp freezers are sent through a bath of cold water, and then frozen into a solid cake, which excludes all air and prevents the oil from becoming rancid.

In this condition, the fish are stored in tiers in the storage rooms at a temperature of 21° F. Generally they are not kept more than 4 to 8 months, although they could be kept in perfect condition considerably longer. They are brought to the storage house in spring and fall, and sold during the season when fresh fish cannot be obtained.

For shipment, the frozen fish are usually packed in barrels. A layer of ice is placed in the bottom, and between every 50 pounds of fish; some ice being also put on top. In order to allow the draining off of the ice water, the barrels should be provided with a sufficient number of holes in the bottom. Shipment is made in refrigerator cars, insuring the arrival of the fish in perfect condition.

The refrigeration necessary for freezing the fish amounts to about 140 B.T.U. per pound, or approximately one ton of fish can be frozen with one ton of refrigeration. Only about one-tenth of this amount of refrigeration is required for the storage of the fish including the cooling of the air of ventilation, which is renewed 2 to 3 times every 24 hours. In other words, approximately 10 tons of the frozen fish can be stored for 24 hours with one ton of refrigeration.

In rooms 10 feet high, about 60 square feet of floor space are required for the storage of one ton of fish, so that one ton of refrigeration is necessary in 24 hours for the cooling of 6,000 cubic feet of storage room space.

Cheese.—During the first 2 to 3 months, cheese should be kept at from 36 to 40° F., the temperature being then gradually reduced to 33° F. and finally to 31° F. This temperature is maintained until the cheese leaves the storage rooms.

A process of paraffining the cheese in order to prevent its becoming moldy has been introduced with good success. The cheese is simply dipped in melted paraffin which covers it with a thin coat impenetrable to air, and which consequently prevents drying of the cheese, and also formation of mold.

No other goods should be kept in the cheese storage room.

One ton of cheese requires 6.6 square feet of space in rooms 10 feet high. Approximately 8,000 cubic feet of space can be kept at the storage temperature by one ton of refrigeration in 24 hours.

Butter.—The temperature at which butter should be kept varies with the length of time it is intended to be stored. For from 3 to 5 months storage, a temperature of 12 to 15° F. is sufficient, but for longer storage, lower temperatures down to about 5° F. are advisable. A recent government investigation showed little difference in effect of temperature between —10° F. and 10° F.

About 8 square feet of floor space are required per ton of butter, and one ton of refrigeration in 24 hours can take care of approximately 4,500 cubic feet of space.

The above remarks apply also to butterine and oleomargarine.

PACKING HOUSE REFRIGERATION.

Refrigeration of packing houses is in many respects, and very naturally, similar to cold storage. However, for cold storage, direct cooling by means of refrigerating coils in the rooms gives very good results, while for packing houses, on account of the nature of the goods handled, this system is not so well adapted.

Cooling Systems.

For the successful preservation of meat, it is not sufficient merely to keep it at a low temperature. The air, which has to be removed frequently, must also be dry, and at the same time as free from micro-organisms as possible. It is evident that the ordinary method of cooling by means of refrigerating coils, which gives such satisfaction in cold storage houses and breweries, is not the ideal system under these conditions. The so-called cold air system, however, serves the purpose admirably.

According to this method the refrigerating device is outside of the cooling rooms, and air which has been cooled to the proper low temperature by this device, and at the same time deprived of its moisture as well as of at least part of the micro-organisms it contained, is circulated through the rooms. In this manner cooling and ventilation are combined.

Two methods of cold air refrigeration can be distinguished: *Wet air cooling,* in which the air comes into direct contact with

cooled brine, and *dry air cooling,* where the air passes a pipe system through which the brine, ammonia or other cooling medium circulates. A number of systems both of the wet and the dry process have been patented and introduced in packing houses, mainly in Europe.

Linde constructed an apparatus in which *rotating discs* dip to about ¼ of their height into brine. The upper part of the discs is, therefore, continuously covered with a thin film of brine which serves to cool the air, which circulates slowly through the interstices between the discs. The air, besides being cooled, is dried, its moisture being absorbed by the brine, and at the same time it is freed from bacteria to a cosiderable extent.

The same manufacturer introduced another apparatus, in which the rotating discs are placed in rows, one row above the other. The brine trickles over the discs and is collected in a gutter, from where a rotating pump again delivers it to the distributing device on top.

In the so-called *rain apparatus,* made by another firm, the air is drawn through the brine, which in the form of a fine spray, drops from a height of about 7 to 8 feet. By this arrangement, the air, on account of its intimate contact with the brine, is absolutely freed from micro-organisms. Furthermore, the apparatus has the advantage of being cheaper both in installation and operation.

The velocity of the air passing through any of the above coolers should be about 120 to 180 feet per minute. In the so-called dry system, the air is drawn through refrigerating chambers where the pipes are arranged in a manner similar to atmospheric condensers. The refrigerant circulates through the pipes.

The moisture of the air is deposited on the pipes in the form of ice, and since an accumulation of the latter seriously reduces the effectiveness of the pipes, its removal at regular intervals is a necessity. In order to permit this without interrupting the cooling process, the chambers are divided into compartments, and as soon as the deposit of ice becomes too heavy, the compartment is shut off and the ice is removed. This can be done in various ways, for instance by circulating the warm gas coming from the compressor through the pipes.

The *fans* for circulating the air should be of such a size that all the air of the refrigerating rooms can be drawn through the chambers in 10 to 15 minutes.

The *heat transmission* of the dry system is approximately 3 B.T.U. per square foot of pipe surface per hour for each degree F. difference in temperature; for the Linde wet air apparatus, the same rule holds good. In the rain apparatus described above, the heat transmission amounts to about 9 B.T.U. per cubic foot of space per hour for 1° F. difference in temperature between the air and the refrigerating medium.

The percentage of moisture in the circulating air in the above systems amounts to about 75%, which has been found most satisfactory. Further reduction of moisture only results in a drying up of the meat.

While the above systems are employed with excellent satisfaction in a large number of European cities, they have found but little recognition so far in this country. Direct cooling of the various rooms by means of direct expansion coils is almost universally employed, and the importance of frequent ventilation and of dry air free from bacteria does not seem to be sufficiently understood, which is the principal reason that the cooling rooms are mostly damp and moldy. There cannot be any doubt, however, that the cold air system will in the future supplant the old method of refrigeration in some industries.

Another cooling method must be mentioned here, as it is similar to the cold air method of refrigeration. According to this system, which has been introduced by the Arctic Ice Company of Cleveland, cooled brine is circulated through the cooling rooms in gutters near the ceiling. As in this manner it comes into intimate contact with the air, the latter, since its moisture is absorbed by the brine, is cooled and dried at the same time. With regard to purifying the air, however, this system can scarcely be so effective as the others described above.

Treatment of Meat.

Meat which is merely to be kept in readiness for the market is stored in rooms which are kept slightly above the freezing point. Freezing it would only spoil the appearance of the meat.

If the air is dry and pure, and if the cold air system is employed, meat may be kept at a temperature slightly above the freezing point for from 6 to 8 weeks.

Such meat as is intended for transport is frozen at a very low temperature and then stored at higher temperatures, but still below the freezing point, until shipment is made. If properly frozen and stored it will keep almost indefinitely.

Chill Rooms.—When coming from the slaughter house, the fresh meat is first put into the so-called chill rooms, where beeves are cooled to about 34 to 36° F., while hogs must be chilled a little more, viz., about 5 degrees lower. The temperature with which the meat arrives at the chill rooms is about 85 to 95° F.

Sharp Freezers.—From the chill rooms the meat enters the sharp freezers, where it is cooled to about 10° F. below zero, in some packing houses even to 20 and 30° F. below. There is, however, no reason for employing such excessively low temperatures, nor in fact for going any lower than 0° F.

Freezing must be done very slowly and carefully, in order to prevent bursting of the meat cells, which would not only spoil the appearance of the meat but would also hasten disintegration after it had been thawed. When the meat is once frozen its temperature can be lowered rapidly. The duration of the freezing process is about 48 hours.

Freezing Process.—When frozen, the meat is placed into the so-called freezing rooms for storage and is kept here at about 20 to 22° F. until it is put on the market.

Thawing of Meat.—For consumption, the meat must be thawed, and this requires the same care as the freezing process. Unless thawed slowly the meat will lose its quality. Several methods for the thawing or defrosting of the meat have been introduced. Usually the meat is put in so-called thawing chambers, through which air of gradually rising temperature is circulated. The temperature of the air at the beginning is 25° F. and this is gradually raised to above the freezing point until a moderately warm temperature is reached.

Space Occupied by Meat.—In chill rooms, about 80% of

the total floor space can be occupied for hanging meat, and approximately 33 pounds of meat can be hung per square foot.

In the freezing rooms only about 70% of the floor space can be utilized for hanging meat, 30 pounds being hung per square foot.

The quantity of meat that can be hung in rooms of given size is easily calculated if it is taken into consideration that the average weight of beeves is 700 pounds, of hogs 250 pounds, of calves 90 pounds, and sheep 75 pounds.

Amount of Refrigeration Required.

For chilling rooms, the amount of refrigeration required amounts to one ton of refrigeration in 24 hours for from 6,000 to 11,000 cubic feet of space, besides the refrigeration necessary for reducing the temperature of the meat. Instead of calculating the refrigeration according to this rule, it may also be found by allowing 3 B.T.U. per 24 hours, per degree F. difference in temperature between the outside air and the temperature of the chill room for each square foot of surface of walls, ceiling and floor, besides the refrigeration required for cooling the meat. The latter calculation is based on the assumption that on the average 3 B.T.U. are transmitted by one square foot in 24 hours through walls, etc., of medium insulation, per degree F. difference between outside and inside temperature.

Assuming, for instance, that a chilling room is 50 feet wide, 60 long and 10 high, and that 140 beeves are entered each day from the slaughtering house, then the refrigeration required in 24 hours can be found by first calculating the heat transmitted by the entire surface:

Floor	50x60.....	= 3,000	square feet.
Ceiling	50x60.....	= 3,000	" "
Side walls.....	60x10x 2.....	= 1,200	" "
End walls.....	50x10x 2.....	= 1,000	" "
	Total =	8,200	" "

Assuming the outside temperature to be 90° F. and the temperature of the chill room 36° F. then the number of B.T.U. transmitted by the entire surface in 24 hours amounts to

$$8{,}200 \times 3 \times (90 - 36) = 1{,}328{,}400 \text{ B.T.U.}$$

To this must be added the refrigeration required for cooling the meat from the temperature at which it arrives at the chilling room to the temperature of the latter. Assuming the temperature of the fresh meat to be 95° F., and the specific heat of meat to be 0.7, then the refrigeration necessary for chilling 140 beeves of 700 pounds each would be:

$$140 \times 700 \times 0.7 \times (95 - 36) = 4{,}047{,}400 \text{ B.T.U.}$$

The total refrigeration required amounts, therefore, to 1,328,400 + 4,047,400 or 5,375,800 B.T.U., corresponding to $\frac{5{,}375{,}800}{288{,}000}$ or 18.7 tons.

It must not be overlooked, however, that the air has to be renewed in the chilling rooms at least every 4 hours, or 6 times a day, which requires additional refrigeration. The cooling of one cubic foot of air of 75% humidity from 90° F. to 36° F. requires approximately 2 B.T.U. in 24 hours, so that the ventilation in 24 hours would necessitate the expenditure of $50 \times 60 \times 10 \times 6 \times 2$ or 360,000 B.T.U., corresponding to a little over one ton. The entire refrigeration for the above chilling room therefore amounted to about 20 tons.

Under ordinary conditions the chilling rooms are not operated at their maximum but only at half their full capacity, which reduces the required refrigeration as follows:

Cooling of meat	2,023,700	B.T.U.
Transmission through walls, etc	1,328,400	"
Ventilation	360,000	"
Total	3,712,100	"

Under ordinary conditions the refrigeration required amounts therefore to:

$$\frac{3{,}712{,}100}{288{,}000} \text{ or } 12.9 \text{ tons in 24 hours.}$$

This corresponds to 120 B.T.U. per cubic foot of space or to 1,200 B.T.U. per square foot of floor space.

For the sharp freezers, the calculations are practically the same as for fish freezing, i. e., approximately one ton of refrigeration is required per ton of meat. Four hundred and fifty B.T.U. may be allowed per cubic foot of space, or 4,500 B.T.U.

per square foot of floor space for rooms of average height, viz., about 10 feet. Beef rooms are usually 15 feet high.

The refrigeration of the freezing room, where the meat is stored, is calculated by adding to the heat transmitted by walls, floor and ceiling, the refrigeration necessary to cool the air for ventilation. In a room of the above size and for an outside temperature of 90° F. and an inside temperature of 20° F., the heat transmitted amounts to:

$$8{,}200 \times 3 \times (90 - 20) = 1{,}722{,}000 \text{ B.T.U.}$$

Since the air must be renewed on the average four times in 24 hours, and since it requires about 3 B.T.U. to cool a cubic foot of this air from 90° F. to 20° F., the total heat brought into the freezing room by ventilation amounts to:

$$30{,}000 \times 3 \times 4 = 360{,}000 \text{ B.T.U.}$$

There must consequently be abstracted from this room in 24 hours:

Heat of transmission	1,722,000 B.T.U.
Heat of ventilation	360,000 B.T.U.
Total	2,082,000 B. T. U.

This corresponds to 70 B.T.U. per cubic foot, or 700 B.T.U. per square foot of floor space.

Amount of Piping Required.

The piping necessary for the proper cooling of the various rooms can easily be calculated if it is remembered that approximately 1,000 B.T.U. are transmitted in 24 hours by one lineal foot of 2-inch pipe. The proper ratio of piping to cubic contents of the rooms would therefore be:

Chill rooms—1 lineal foot of 2-inch pipe to 14 cubic feet.

Sharp freezers—1 lineal foot of 2-inch pipe to 8 cubic feet.

Storage rooms—1 lineal foot of 2-inch pipe to 22 cubic feet.

MARINE REFRIGERATION.

In former years, refrigeration on board ship was produced by air machines exclusively, these being preferred for reasons of safety. At the present time, however, ammonia and carbon dioxide compression machines are mostly employed, and the lattel are generally given the preference. For the same amount of refrigeration produced, the carbon dioxide compressor is much

the smaller. Carbon dioxide itself is absolutely neutral towards metals, which allows the employment of copper in the construction of the machine, the use of this metal being desirable on account of its flexibility and resistance against the action of water. Furthermore, carbon dioxide will extinquish a fire, and is less dangerous to life than ammonia, should a large quantity escape on account of a break.

On the other hand, carbon dioxide machines are not so well suited for ship service in tropical climates, where the temperature of the water that must be employed for cooling is often between 80 and 90 degrees F. With such high cooling water temperatures, the performance and the efficiency of this machine are considerably reduced. The ammonia machine, however, is comparatively little affected under such circumstances.

Refrigeration on board steamers is especially valuable for the transportation of fresh meat, which is shipped from South America or Australia, for instance, in enormous quantities.

In rooms 6 feet 8 inches high, about 60 pounds of beef can be hung per square foot; if the meat is packed instead of hung, about 140 pounds can be stored per square foot. The former method, however, is preferable, because it allows free circulation of cold air, which is necessary for satisfactory preservation of the meat.

Freezing and storing temperatures of meat will be found under "Packing House Refrigeration."

REFRIGERATION USED FOR VARIOUS PURPOSES.

Sinking of Shafts Through Quick Sand.

Very satisfactory results have been obtained in the sinking of shafts through strata of wet sand by freezing the soil around the contemplated shaft. For this purpose a sufficient number of holes are bored vertically into the ground so that they form a circle of the proper diameter, and iron pipes are inserted. By a suitable arrangement, a refrigerating medium is circulated through the pipe. The wet soil gradually freezes around the pipes, and a solid wall is formed.

Usually brine is employed for circulating through the pipes,

but there is no reason why the pipes could not be cooled by direct expansion as well.

In this device, the brine enters a distributing pipe, from where it flows down the small tube which is placed inside the freezing pipe. The latter is closed at the bottom and the brine must consequently rise in the annular space between inner tube and wall of freezing pipe until it reaches the return ring, from where it is led back to the refrigerator in order to be again lowered in temperature.

The brine enters the pipes at 0° F. and leaves them at 7° F. Its velocity in the descending tube is usually about 275 feet per minute, while it rises in the pipe with a velocity of 25 feet per minute.

One square foot of pipe surface transmits about 90 B.T.U. per hour. When sinking a shaft it is of the utmost importance that the freezing tubes penetrate far enough into the underlying solid strata to sufficiently support the wedging crib. It is furthermore essential that the bore holes for the freezing pipes be as nearly vertical as possible and that the freezing tubes work with uniformity; otherwise the ice wall which forms around the tubes will not be of equal thickness and is liable to collapse, not being able to resist the outside pressure.

The thickness of the ice from the pipes towards the outside is about 18 to 20 inches, while it amounts to 36 to 40 inches towards the inside, making the total thickness about 4½ feet.

To sink a shaft of a diameter of 15 feet to a depth of 165 feet would require about 20 pipes, arranged in a circle 20 feet in diameter. A solid ice wall of 14 feet inside and 23 feet outside diameter and having a volume of about 43,000 cubic feet is formed during the freezing process. Assuming that the soil consists of 80% of solid matter and 20% of water, one cubic foot would weigh about 146 pounds, making the total weight of the soil to be frozen approximately 6,300,000 pounds.

From this mass, about 50 B.T.U. must be abstracted per pound, including overhead losses, so that the total abstraction of heat necessary amounts to $6{,}300{,}000 \times 50$ or 315,000,000 B.T.U.

Assuming that 3-inch freezing pipes are employed, the total pipe surface would be 3,600 square feet. As each square foot transmits about 90 B.T.U. per hour, it would require $\frac{315,000,000}{3,600 \times 90}$ or 972 hours or 40½ days to freeze the protecting wall around the shaft.

The capacity of the refrigerating machine would have to be $\frac{315,000,000 \times 24}{972}$ or 7,778,000 B.T.U. per 24 hours, which corresponds approximately to a 30-ton machine.

Refrigerator Cars.

Refrigerator cars are cooled with ice, there being an ice chamber on top of the car. The warm air flows over the ice and after this is cooled sinks into the car. If the insulation is good, the ice consumption varies from 300 to 700 pounds in 24 hours.

There is no reason why artificial refrigeration could not replace cooling with ice. Since comparatively little refrigeration is required, a small refrigerating machine would answer the purpose.

Skating Rinks.

In former years, brine pipes, laid directly on a watertight floor, were covered with water and the latter frozen. In order to keep the surface smooth the rink had to be flooded from time to time and a new surface formed by freezing. At the present time skating rinks are made in a different manner. A large, shallow reservoir containing the water to be frozen is placed into another slightly larger reservoir which is filled with brine. The inner reservoir floats on the surface of the brine which serves to freeze the water. This arrangement is superior to the method formerly employed, in that such floating skating rinks are more elastic.

If every part is properly insulated, the machine must be able to abstract 100 B.T.U. per square foot per hour, and calculations can be made accordingly.

About 2 lineal feet of 2-inch piping are required per square foot of ice.

A temperature of 23° F. has been found by experience to

Ring Armatures.—The coils are wound around a core in the form of a ring. The end of one coil is connected to the beginning of the next coil, and consequently the coils are in series. A short wire connects the junction of two coils with a commutator segment, of which there are as many as there are coils on the armature ring.

Drum Armatures.—In these armatures, the coils are wound longitudinally upon the core, and each coil is connected to one of the commutator sections. In the ring armatures, only the wire on the outside of the coil cuts the lines of force, and the wire on the ends and inside of the coil is "dead wire," and only serves to conduct the electricity. In the drum armature, both sides of the coil cut the lines of force, and only the ends of the loop are inactive. The field magnet must be magnetized either by the dynamo itself or some outside source.

Residual Magnetism.

After the field magnet has once been magnetized or excited from any source of electricity, for instance, from a direct current dynamo, the fields always retain some magnetism. This is called *residual magnetism*, and it means that there are always some magnetic lines of force crossing from one pole to the other.

Electromotive Force.

When the armature begins to rotate between the north and south poles of the field magnet, there will be lines of force induced in the armature coils. The action of the electromotive force causes electric currents of a weak potential to be generated in the armature coils. These currents are collected from rings, as in the case of alternating current, or from a commutator by suitable brushes, as is necessary with direct current. They are sent through the wire, which is wound around the field, and hence a higher potential is generated. This soon raises the current to its normal predetermined strength, at which it enters the outside circuit, where it can do actual work by overcoming the resistance interposed for this purpose, in the form of producing light, driving electric motors, etc.

The *induction* of the electromotive force (commonly abbreviated E. M. F.), in the armature coils, takes place as follows: When one of the coils of the armature approaches the north pole of the magnet, a current or E. M. F. is set up, or induced in this

coil, which E. M. F. increases in strength as it comes nearer to the pole, and reaches its maximum strength when the coil is directly opposite the middle of the pole; then, as the coil moves away from the pole, the E. M. F. gradually decreases in strength, and ceases to flow when the coil is at a point equidistantly removed from the north and south poles of the magnet. When this point is passed, and the coil begins to move towards the south pole, a current is again set up in the coil, but this time it flows in the opposite direction, and as before, gradually increases and decreases in strength from a maximum to a minimum as it approaches or recedes from the respective pole.

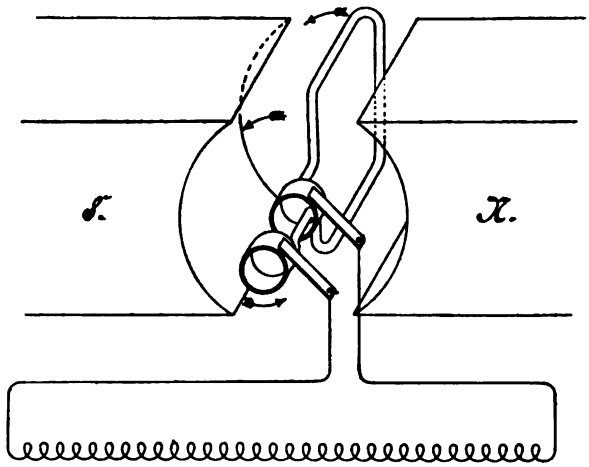


Figure 232. Elementary Alternating Current Dynamo.

There are then, for every rotation of a coil, two changes in the direction of the electric current or two *alternations*, and this is called a *cycle*. The number of cycles that occur in one second is termed the *frequency*.

It is well to keep these terms in mind, as they are of especial importance in all alternating current systems.

The number of poles P is found by multiplying the frequency F by 120 and dividing the result by the revolutions per minute R, i. e.,

$$\text{Number of poles } P = \frac{F \times 120}{R}$$

The frequency or number of cycles F is found by the following formula:

$$F = \frac{P \times R}{120}$$

Example.—If $P = 8$ poles in the machine, and $R = 900$ revolutions per minute, what is the frequency or number of cycles of this machine per minute?

Solution.—$F = \frac{8 \times 900}{120} = 60$ cycles per minute, which are equal to 7,200 alternations per minute.

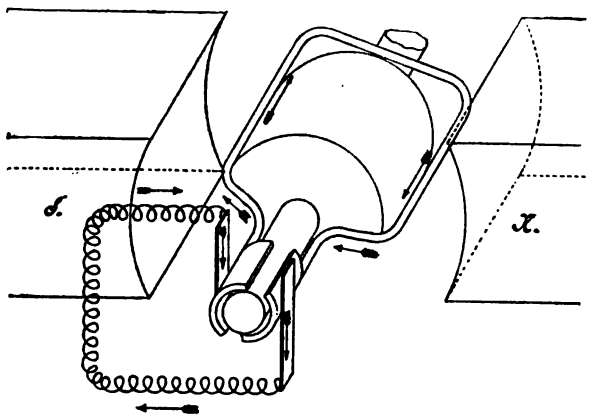


Figure 233. Elementary Direct Current Dynamo.

In order to make the current flow in one direction only, i. e., if the positive current wire is to remain positive and the negative remain negative, a *commutator* is placed on the end of the armature. The term *commutating* means to change, and for the subject treated here would mean to change an alternating current into a continuous current, for which purpose a commutator is used. It consists of two or in fact any number of copper segments insulated from each other. Two or more copper or carbon brushes press lightly against the segments, for the purpose of making electrical contact only. Figure 233 shows how the alternating current is made to flow in the same direction in the outer circuit. The strength of the current flowing in the outer circuit is, however, by no means constant. In the case of a two-part commutator it rises from zero to the maximum and falls again to zero twice in every

revolution, but always in the same direction. These fluctuations or pulsations in the strength of the current give it the name of *pulsating current*. In order to make the pulsations of current less noticeable and almost constant in value, a considerable number of coils are used and connected to as many commutator segments, the planes of the coils being placed at equal angles to one another.

The strength of the current in E. M. F. depends upon the number of conductors (coils), the method of connecting them, the speed of rotation, and the strength of the magnetic field. To form an idea of how it is possible to create an electric current in the copper wire which forms the coils, by simply rotating the wire between the poles of the magnet, we may consider the magnetic field between the poles of a magnet as the place of two ether storms blowing in opposite directions from each pole, and neutralizing each other and producing a calm in the center plane between the poles. When the rotating wire dips into this storm, the ether upon the surface of the wire is compressed, *magnetic whirls* are set up around that part of the wire which is in the magnetic field exposed to the storm, and the thus totally strained ether reacts upon the ether along the wire in the form of an electric wave or current. Mechanical energy must be spent to drive the wire against the storm, much the same as moving any substance against resistance. The work done upon the compressed ether in the dynamo, however, is utilized in the form of an electric current, for producing light, power, etc.

Armature Reactions.

The electric current of the armature also reacts upon the core, and turns it into a magnet, having its north pole at the negative brush and its south pole at the positive brush. These poles of the armature core again react upon the magnetic lines of the field, and as a consequence the neutral points are moved from the central plane between the two poles of the field magnet in the direction of the armature rotation. The brushes must therefore be rocked forward until they strike the neutral point, or until the sparking ceases, or very nearly so. The stronger the current, the farther the brushes must advance. In modern, well-designed machines, the aim is to reduce this shifting of brushes as much as possible in regard to change of load, and in some no shifting is necessary at all.

Hysteresis.

The iron core of an armature, when revolving between the two poles of a two-pole field, will be alternately magnetized and de-magnetized in each revolution. The magnetization of the iron consists in turning all the molecules to the lines of the magnetic field, just like so many vanes in the wind. When this action of magnetization and de-magnetization takes place, it has been discovered, that the iron has a strong tendency to retain, as it were, residual magnetism, even after the magnetizing force has been removed. This tendency to cling to the preceding condition of magnetization is called *hysteresis,* and, as it causes heating of the armature core, must be considered a loss, as a part of the mechanical energy driving the armature is thus changed into heat and lost.

Eddy Currents.

The iron core of the armature is not only a conductor of magnetism, but like all metals, also of electricity, and when rotated in the magnetic field, electric currents are produced in the core as well as in the coils. However, these currents in the core are not transmitted to the external circuit, they flow around in the core, heating it, as well as the copper wire in the coils, thereby increasing the resistance of the wire. These currents in the core are called *eddy currents.* In order to weaken them, the armature is built of thin sheets of iron *(laminated core).*

Field Magnets.

The field magnet of a dynamo consists of either two, four or six, and so on, iron cores, usually called *pole pieces,* around which insulated copper wire is wound. The cores are generally made of cast iron, though wrought iron would be better, but owing to the cost of working it into shape, the former has been found more economical. The space between the pole pieces is the *magnetic field,* so called because it is there that the magnetic lines of force cross from one pole to the other. The armature revolves in this field as close to the pole pieces as mechanically possible. The spaces between the armature and the pole pieces are called the *air-gaps.*

Classification of Dynamos.

The different dynamos the engineer comes in contact with are mainly the direct current constant potential dynamo, and in some

instances the constant current dynamo, and for large establishments the alternating current dynamo. There are, however, a large number of other dynamos, constructed for various purposes, but only the above-named will be treated here.

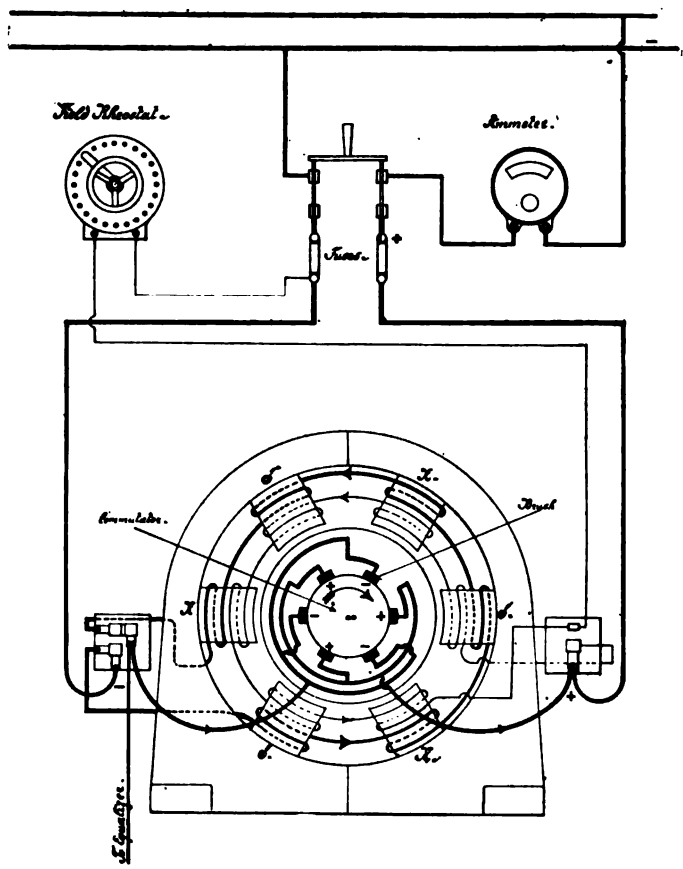


Figure 234. Compound Wound Dynamo.

Direct Current Constant Potential Dynamo (Compound Wound).

The fields of a constant potential dynamo are usually compound wound, which means that the main wiring, consisting of small size wire, is shunted from the external circuit *(shunt winding)*, and the other wiring, less turns than the former, con-

sisting of heavy wire in series with the external circuit *(series winding)*. Through the latter, the entire circuit passes at all times, whereas through the former, a part only of the current is allowed to pass. A regulator is connected in this wire in series, the so-called *rheostat,* consisting of a number of resistance coils,

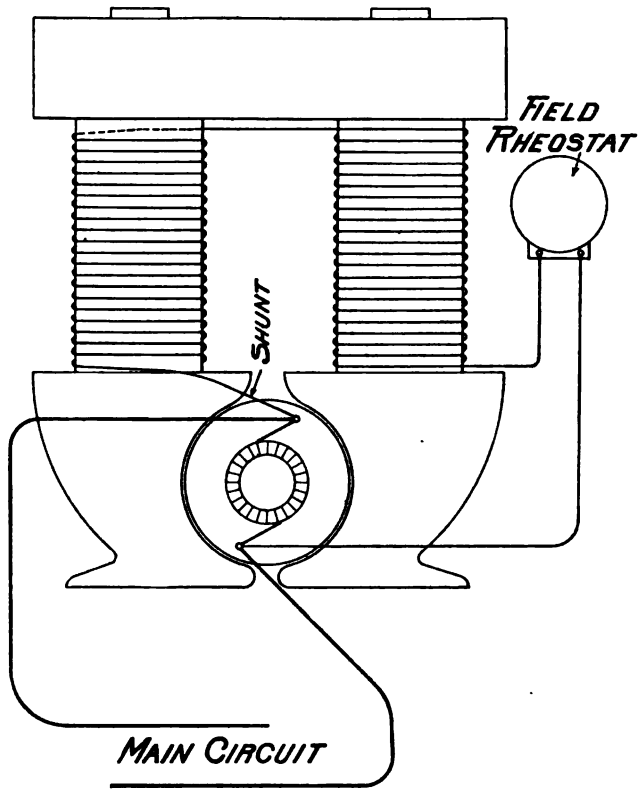


Figure 235. Shunt Wound Dynamo, Two Poles.

and by letting the shunted part of the current pass through either more or less of these resistance coils, the E. M. F. is either increased or decreased at the terminals. The series winding around the pole pieces serves the purpose of decreasing or increasing the field magnetism, according to the variation of the load. If this winding were not present, the E. M. F. of the dynamo would vary every time the load varied, and a regulation by hand on the rheostat would be required at every fluctuation of the load. A com-

pound wound dynamo combines in its fields the winding of both the series and shunt machines. They may be wound to keep a constant electromotive force at varying loads, or may be over-compounded so as to allow for loss in feeders, as each specific

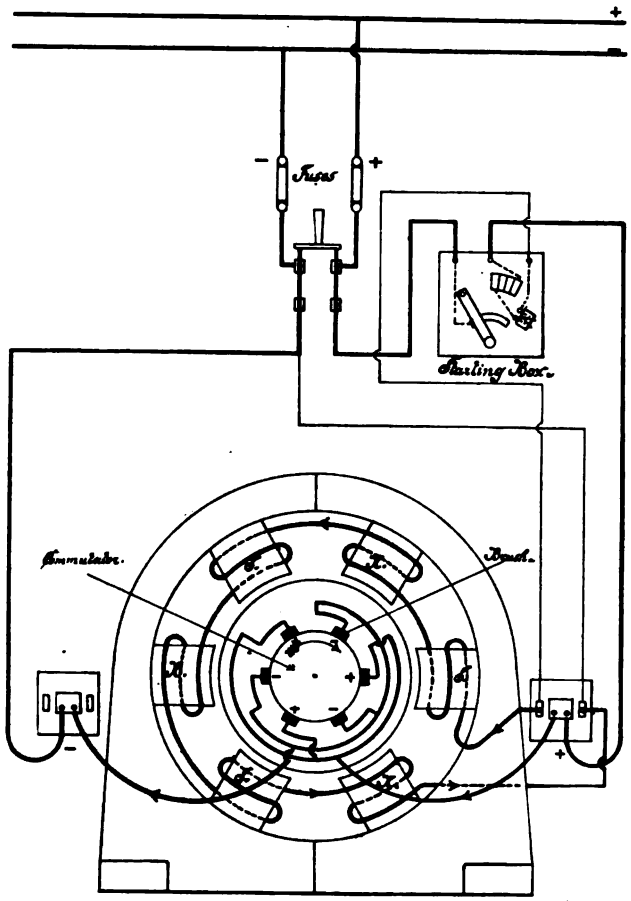


Figure 236. Shunt Wound Dynamo, Six Poles.

case may require. Figure 234 shows a graphical sketch of a compound wound modern dynamo.

Constant Potential Shunt Wound Dynamos.

With a constant potential (shunt or compound wound) dynamo, the voltage remains practically constant and the current

strength varies with the load. Their use is general for incandescent lighting and power circuits, being constructed as a rule as multi-polar machines for this purpose.

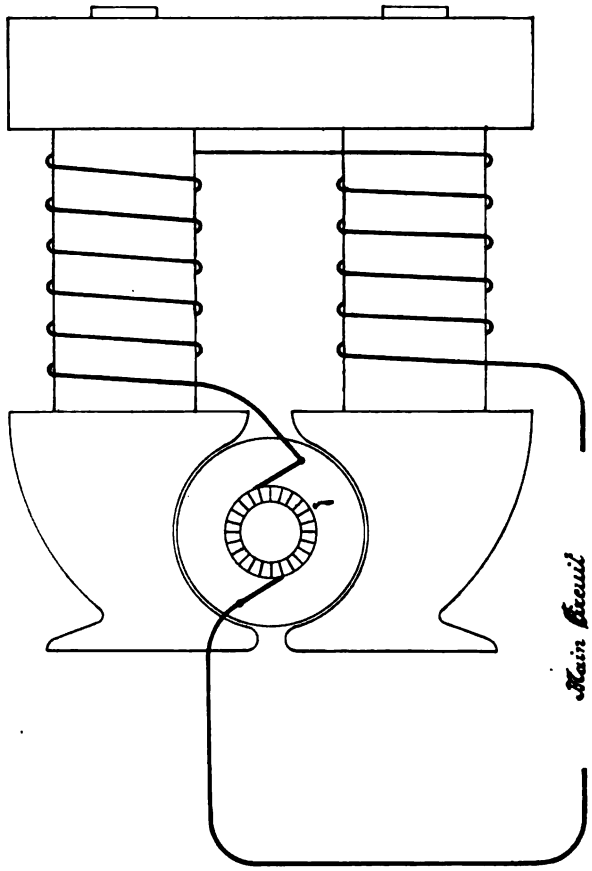


Figure237. Series Wound Dynamo.

Constant Current Series Wound Dynamos.

In a series wound dynamo, all of the current passes through the field-coils before passing into the external circuit and back to the negative brush. These dynamos are usually called *constant current dynamos,* because the current strength (amperes) remains constant, while the voltage varies with the load. They are

commonly used for arc lighting of long distance distribution, such arc-dynamos generally being of the bi-polar type.

Field Rheostat.

With the constant potential dynamo, the voltage in the external circuit is regulated by varying the current through the field coils by means of a variable resistance called a *field rheostat.* The current flowing through the shunt is equal to the potential difference across the brushes divided by the resistance of the shunt itself and the resistance of the rheostat. The rheostat used for this purpose is connected in series with the shunt field winding. It is usually built of a number of resistance coils, arranged in such a way that by means of a sliding contact the shunted current can be made to flow through none or all. The sole object of a rheostat for this purpose is to vary at will the strength of the field magnet and thus the E. M. F. generated by the armature. Strictly speaking, it regulates the electro-magnetic induction by either increasing or decreasing the strength of the magnetic poles, whereby the E. M. F. is kept constant, providing the resistance in the outer circuit remains constant. It is evident that if the load varies, the resistance varies, according to Ohm's law, $E = I \times R$. If, for instance, the armature circuit, which should deliver 100 amperes, has a resistance of 0.03 of an ohm, then it would require $100 \times 0.03 = 3$ volts, which would be used up in driving the 100 amperes through the armature circuit. If the resistance in the outer circuit is 0.06 of an ohm, and the 100 amperes should be delivered at the end of distribution with an E. M. F. of 110 volts, then the loss in the outer circuit would be $100 \times 0.06 = 6$ volts, plus 3 volts lost in the armature circuit or 9 volts in all. This loss is called *drop.* The available voltage at the end of distribution would thus be 9 volts less than the current generated in the armature proper; hence if 110 volts are wanted at the end of distribution, the armature must generate 119 volts.

In order to make up for any sudden load variations, the field magnet is compound wound as previously stated, and if properly designed, no regulation of the rheostat should be required with medium variations of the load. The ratio of the rise of voltage from no load to full load condition should be predetermined and fixed for each particular machine, and should adjust

itself automatically, without necessitating the touching of the field rheostat.

Brushes.

For collecting the current, brushes made of copper or carbon are used. On account of the small cost, the latter kind is principally used, but to get the same resistance of contact, they are made much thicker and wider than copper brushes under like conditions. The thickness of a brush should not exceed two or three times the width of one commutator segment. The contact area of a copper brush should not be less than one square inch per 250 amperes, and for a carbon brush not less than one square inch for 37½ amperes. One or more brushes are used side by side, if necessary, in order to obtain the desired contact area. In order to wear the commutator surface evenly, they should be set staggered. The brush-holder is usually made so it can be shifted to a non-sparking position.

E. M. F. and Magnetic Flux.

The design of a dynamo involves many accurate calculations, but the fundamental principle consists in ascertaining the useful magnetic flux equal to the E. M. F., and may be expressed in the following formula:

$$E = \frac{F \times N \times R}{10^8 \times 60}$$

E = volts generated by the dynamo, called the potential.

F = number of lines of force passing from the north pole through the armature to the south pole.

N = the number of conductors in a bi-polar machine on the surface of the armature (they may be in one or more layers), or in a multi-polar machine, as measured from a point opposite one north pole to a point opposite the next succeeding north pole.

R = the number of revolutions the armature should make per minute.

Example.—In a bi-polar machine with 250 conductors on the surface of the armature, the latter having a speed of 1,200 revolutions per minute, and the lines of force passing from pole to pole being 2,500,000, what would be the electromotive force generated?

Solution.—$E = \frac{2{,}500{,}000 \times 250 \times 1{,}200}{100{,}000{,}000 \times 60} = 125$ volts.

If this were a four-pole machine, the number of conductors and lines of force being the same as in the previous example, then the speed could be reduced one-half, or the voltage could be doubled, or the number of conductors could be reduced one-half, or the lines of force could be reduced one-half.

The lines of force must pass through a circuit of three paths, namely, the field magnet, the air-gap twice, or rather through a double air-gap, and through the armature core. A given density of force should not be exceeded, and this amounts to about 50,000, 90,000 and 110,000 lines of force per square inch for ordinary cast iron, wrought iron forgings or cast steel, and for soft sheet iron respectively. Since the *permeability* (ratio of magnetization to magnetizing force) of air is very low, the air-gap is usually made very small. The size of wire on the armature, and the manner of connecting the winding, determine the amount of current the dynamo can give, since the larger the wire the more amperes can pass through without overheating, the limit being about 600 circular mils per ampere. The heating effect of the current on conductors of the armature is subject to numerous conditions, and it is almost impossible to formulate a rule of any value. In stationary conductors, such as the fields, etc., the rise in temperature is about 200 times the number of watts divided by the radiating surface in square inches. The manner of connecting the winding affects the amount of current when the number of paths is increased, as for instance, a four-pole machine with parallel winding will allow twice as much current to pass through the armature as a bi-polar machine, since there are four paths in the former and only two paths in the latter.

Torque of an Armature.

By the torque of an armature is meant the moment tending to turn it, or the pull at the surface of the armature multiplied by its radius. In a dynamo, it is the turning moment which must be applied to the armature in order to turn it against the opposing electro-magnetic force. In a motor, it is the turning moment the armature gives off to overcome external resistance. The torque is generally expressed in pound-feet, which is equal to the pull in pounds, times the leverage in feet.

Example.—What is the torque of an armature which has

a pull of 1,200 pounds at the surface, and which has a radius of 8 inches ($^8/_{12}$ foot)?

Solution.— $\frac{1{,}200 \times 8}{12} = 800$, which is the torque of this armature, or 800 pound-feet. There is a difference between pound-feet and foot-pounds, as the latter means the pull in pounds times distance of travel. The two expressions are, however, proportionate to each other, but the ratio varies with the speed. If the torque and revolutions per minute of an armature are known, the foot-pounds can be calculated by the following formula:

$$\text{Foot-pounds per minute} = T \times 2 \times 3.1416 \times s.$$

T = torque in pound-feet.

s = revolutions of armature per minute.

Assuming the revolutions of the armature to be 750 per minute in the above example, the foot-pounds per minute of the armature would be:

$$800 \times 2 \times 3.1416 \times 750 = 3{,}770{,}000 \text{ foot-pounds.}$$

Expressed in horse-power, it would be:

$$\frac{800 \times 2 \times 3.1416 \times 750}{33{,}000} = 114.2 \text{ H. P.}$$

Efficiency of Dynamos.

The efficiency of a dynamo depends upon its speed, construction, capacity, and various other conditions; in fact, two dynamos of the same construction and capacity seldom show exactly the same efficiencies. The principal losses are:

Mechanical friction.....1½% to 5%.
Core losses.....1 % to 2%.
Field losses.....1½% to 10%.
Armature losses.....1½% to 12%.

The average efficiency of dynamos at full load may be taken as 90% for medium sizes, ranging less for ¼, ½ and ¾ load conditions.

Direct Current Motors.

An electric motor is the reverse of a dynamo, and is a machine that converts electrical power supplied to it into mechanical power. Conversely, any motor driven by mechanical power will deliver electrical energy.

Direct current motors are built practically the same as dynamos, but the efficiency is usually less than that of dynamos.

The average efficiency of motors is about 80 to 85%, and if not exactly known, may be assumed as such. For direct current there are three kinds of motors in general use, namely the shunt-wound, the series-wound, and the compound-wound motor. The shunt and compound types are most generally in use for stationary work, while for transportation purposes, i. e., trolley cars and hoisting machinery, the series motor is most extensively applied.

Counter-Electromotive Force of a Motor.

Since the armature of a motor is made to rotate in a magnetic field, it cuts the magnetic lines of force just as in a dynamo, and consequently an electromotive force is induced in it, which acts in opposition to the electromotive force applied to the motor, and is therefore called the counter-electromotive force of the motor.

The faster the motor armature runs, the greater does this counter-electromotive force become, and when the motor runs without the load, it is almost equal to the electromotive force, and therefore the motor takes but very little of the current. The counter-electromotive force at a certain speed is equal to the electromotive force which the motor would develop if run as a generator at this same speed. The current through the armature of a motor is found by the simple motor equation, that is by dividing the difference between the applied electromotive force and the counter-electromotive force by the resistance of the armature. If the E. M. F. from the dynamo is E volts, the counter E. M. F. e volts, and the resistance of the motor armature r ohms, then the current I in the motor armature in amperes is

$$I = \frac{E - e}{r}$$

or by transposition,

$$I \times r = E - e.$$

$$e = E - I \times r.$$

The last equation shows that the counter-electromotive force is always less than the E. M. F. of the dynamo by an amount equal to the volts-drop ($I \times r$) in the armature.

Example.—A motor is connected to a 110-volt circuit, the resistance of the motor armature being 0.1 of an ohm. What is the current flowing through the armature when the motor is standing still?

Solution.—Current $= \frac{110}{0.1} = 1{,}100$ amperes.

The armature of the motor now begins to rotate, and consequently the current through the motor armature is gradually checked, as the counter E. M. F. developed by the rotation increases.

Example.—Suppose that at the desired speed this counter E. M. F. amounts to 95 volts, what is then the current supplied to the motor?

Solution.—Current $= \frac{E - e}{r} = \frac{110 - 95}{0.1} = \frac{15}{0.1} = 150$ amperes, which would be $150 \times 110 = 16{,}500$ watts.

The watts developed by the motor would be:

$150 \times (110 - 15) = 150 \times 95 = 14{,}250$ watts. Hence the horse-power given off by the motor would be:

$$\frac{14{,}250}{746} = 19.1 \text{ H. P.}$$

Example.—What would be the efficiency of this motor in per cent?

Solution.—Efficiency $= \frac{14{,}250 \times 100}{16{,}500} = 86.4\%$.

When the load of the motor is increased, the speed begins to slacken, and the counter E. M. F. decreases, and the motor can take more current. The counter E. M. F. thus regulates the supply of electrical energy, acting as a resistance without the disadvantage of turning a part of the electrical energy into useless heat.

Starting Boxes and Speed Regulators.

In starting a direct current motor of any kind, especially a shunt wound motor, it becomes necessary to check the current slowly, by means of a starting box, permitting the armature to gradually reach the full speed while the field resistance is being cut out. In order to do this, a series of resistance coils are connected with the armature circuit, and so arranged that the flow of current can be gradually decreased as the speed of the armature increases. This device is called a starting box. It is similar in construction to a rheostat. A starting box should, however, not be used as a speed regulator, for although it can reduce the

speed for short periods to some extent, it is not advisable to do so, as it may result in over-heating the resistance coils and causing injury.

There are a great many varieties of starting boxes and speed regulators on the market.

The former are often equipped with either overload or no-voltage release, or even possess both features. Such starting boxes will release the lever and open the circuit in case the load becomes excessive, thereby protecting the motor. The no-voltage feature likewise is a protection, as it will cause the circuit to open at the starter, should the line voltage for any reason drop momentarily.

The speed regulators or controllers are made in three distinct types; and the changes of speed on direct current motors can be arranged either by *field resistance,* which allows a limited range of increase above the normal speed within about 20%; or *armature resistance,* which is inserted into the armature circuit for reducing the speed of the motor to any degree within about 50% of the normal. Armature resistance cannot be inserted without a corresponding loss of energy. The third type combines field resistance and armature resistance in one regulator, where a great range of speed is desired.

All the above described starting devices must be operated by hand. However, as it is often the case that control is desired from a distance, as for instance from the operator of a platform elevator or from mechanical devices such as float balls in tanks, etc., the *automatic starter* has been perfected with great success.

Operation and Care of Motors.

The general care that dynamos and motors should receive may be expressed as follows:

Keep the commutator smooth and round, the brushes in good order so that they make good contact with the commutator and at the right points. Keep the dynamos and motors clean and well insulated from the foundation. Keep the circuits in good order, keep the lines as rigid as possible and the insulation resistance as high as possible. Keep the dynamos and motors well protected by the proper size and kind of fuses or circuit breakers. Keep water away from all electric devices.

Lubrication of bearings must be well attended to, and it is

important that no oil should find its way into the armature or field coils of the motors.

An excellent way to clean motors and generators is by means of compressed air.

Never use emery cloth on the commutator, as the fine loose particles will eventually short-circuit the segments. Use even sand-paper sparingly and with judgment. The surface should be glossy and of even color. A motor of good design and workmanship should not spark.

Sparking troubles are more common and more annoying than many other defects. Sparking may be due to faulty construction, but trouble from this cause in modern designs is very remote. Various reasons for sparking may be accounted for even in good motors.

As mentioned previously, good contact of the brushes is all-important, and any noise from chattering of brushes should be promptly attended to. The heat produced by poor contact often brings the commutator to such a high temperature that a blackened, rough surface as well as melting of the soldered joints is inevitable.

Overload conditions of a motor and wrong position of the brushes will also produce sparking.

Loose segments and "high points" on the commutator should be investigated and remedied.

If the brushes have become coated with copper on their surface next to the commutator, sand-paper should be applied, rough side towards the brush and between brush and commutator, until they are ground perfectly clean and true to the commutator surface.

A piece of ordinary grind-stone trimmed to the curvature of the commutator can also be used for this work with good effect.

Heavy and long sparks usually indicate loose connections of commutator bars, and such should be re-soldered.

Poorly fitting brushes are often the cause of reduced speed below the normal, due to the drop of voltage in the brush itself.

Badly worn commutators should of course be turned to an even, smooth surface in a lathe. High mica is frequently found to be the cause of sparking and heating, and it may become

necessary to carefully cut out some of the mica between the slots to a depth of about $^1/_{16}$ of an inch with a chisel.

Lubricating the surface or using "compounds" are only for the temporary relief of sparking.

Ordinarily a brush should cover only two segments at a time.

Should a direct current motor be running in the wrong direction, all that is necessary is to reverse the field connection.

Testing of Direct Current Dynamos and Motors.

The most vital points of efficiency and general satisfaction in dynamos and motors are closely related to the temperature of the working parts under operating conditions; the maxim holding true also in this case, that where heat is produced and wasted, energy must be lost. Besides that, excessive heating must in time affect the material and destroy the rubber insulation by slowly, but surely, carbonizing it.

Permissible limits of temperature have therefore been agreed upon by the American Institute of Electrical Engineers, giving as standard temperatures for the above type of machines 50 degrees Centigrade rise by resistance in field or armature, 55° C. rise by thermometer in the commutator and 40° C. rise by thermometer in the bearings. Rise of temperature means the actual temperature observed on the parts in question minus the observed room temperature.

The above rated temperatures are based on being taken after a continuous full load 10 hour run, and if after the completion of this run 25% overload is carried for 2 more hours, the rise in temperature should not exceed 15° C. A good machine should even stand short periods of 50% overloads, and shifting of brushes should not be necessary from no load to full load.

These readings can easily be taken by the engineer, who must provide himself with two or three so-called "Chemical" Centigrade thermometers, which are simply attached to the parts to be tested by common putty.

Should the full load or overload condition not be available at the time when the generator is to be tested, a water rheostat can be employed for the purpose. The latter consists of a wooden tank containing about 15 cubic feet of water per 125 K. W. Two iron plates, properly connected to the generator, are suspended

The rubbing or circumferential speed $(V) = 2r_{\pi}n$; $n =$ the revolutions per minute which the machine is able to keep up steadily.

The energy consumed by the friction in foot-pounds is equal to the output of the engine in foot-pounds.

$$E = W \times V = \frac{1}{r} \times Q \times 2_{\pi}r \times n$$

$$E = 2_{\pi} \times l \times Q \times n$$

foot-pounds per minute, or

$$\text{H. P.} = \frac{2 \times 3.1416 \times 1 \times Q \times n}{33{,}000}$$

If the length of the beam (l) is made 5¼ feet, the expression $\frac{2 \times 3.1416 \times l}{33}$ cancels and we get

$$\text{H. P.} = \frac{Q \times n}{1{,}000}$$

(The influence of the weight of the beam may safely be neglected).

Example.—How much is the output of a machine running at 200 revolutions per minute with $Q = 50$ pounds weight in the scale pan of above brake?

Solution.—Output $= \frac{50 \times 200}{1{,}000} = 10$, which is the brake horse-power of the machine.

The braking pulley must be made large enough, otherwise the friction, on account of the very high pressure, will fluctuate, and it is not possible to hold the beam steadily in equilibrium.

The heat produced by the friction has to be carried off by a stream of cold water.

Fuses.

A fuse is a piece of leaden wire (lead with a small percentage of tin), which is inserted in series with the circuit. When the current becomes excessive the heat produced thereby melts the fuse and thus opens the circuit. In other words, the fuse is the safety valve of the dynamo, protecting it from injury.

The carrying capacity of a fuse is the number of amperes it can carry without melting or "blowing." In motor circuits the highest power in watts consumed by the motors at the lowest possible voltage should be the guidance for selecting the proper size of fuse. It should be borne in mind that the amperes for a given number of watts increase as the voltage decreases, as expressed by the formula:

$$I = \frac{W}{E}, \text{ or amperes} = \frac{\text{Watts}}{\text{Volts}}$$

Fuses must be installed wherever a change in size of wire takes place, that is, where a smaller wire leads off from a heavier one. All switches must be protected by fuses. The *open link fuses* are now being generally replaced by *enclosed fuses,* the latter being fireproof, on account of being confined into a solid shell filled with asbestos.

Whenever a fuse "blows" it should be the first duty to look for the cause, which may either be a ground on the machine, overload, too much friction due to lack of lubrication, or a break or foreign obstruction within the driven machine or motor.

Fuses should not be used heavier than 25% above the necessary required current.

For larger motors and on switchboards and generators *circuit-breakers* are employed. They operate with great accuracy, can be adjusted to the load within certain limits, save the expense of large fuses blowing, and can quickly be "thrown in" again to close the circuit, without causing delays.

Switchboards.

The most important accessory to the dynamo-electric machine is the switchboard, as it contains the necessary switches, safety devices and instruments for the guidance of the engineer. It should contain besides the necessary generator main switches, fuses and circuit-breakers, at least a voltmeter, an ammeter for each generator connected thereto and ground detector, usually mounted on the so-called generator panel. The current from the main switches is transmitted by heavy copper "bus bars" to the so-called "feeder panels," which in equipment resemble the former.

To run generators in parallel, equalizing connections and switches are necessary, and these may be mounted also on the

switchboard, although in some cases individual switch stands are employed.

Desirable instruments for closer observation of the output, as well as for the keeping of records, are wattmeters, these being usually installed on the feeder lines. For the checking up of the total current produced by any generator *integrating wattmeters* are used, while for graphic records of the voltage and amperes, *recording voltmeters* and *recording ammeters* are in great demand.

The construction and design of the switchboard must be adapted to each particular condition, kind of current, system of wiring, etc.

In general it should be well lighted, kept clean and placed in dry surroundings, accessible both in front and back. All bolted and clamped connections should be kept drawn up tight to prevent heating. All instruments, like gauges, should be frequently tested as to their accuracy.

Alternators.

A dynamo which generates alternating currents is called an alternator, and this type of machine is usually made multipolar. The frequency required in practice varies from 20 to 140 cycles per second; however, 25 and 60 cycles are in most common use. A bi-polar machine which should give 60 cycles would have to make 3,600 revolutions per minute and would have to be driven by a steam turbine. The armature in large alternators is stationary and the field revolves, and in small machines the reverse is the case. Alternators are built either single, two or three phase, which means that either one, two or three different electromotive forces rise and fall in different intervals. Three-phase machines seem to give the best results. The field magnets of an alternator are separately excited by a separate direct current dynamo, usually 125 volts, which is called the *exciter*. Alternators are generally built to generate 1,100 to 2,200 and more volts, but for lower voltage, machines are also designed, or the current can be transformed to 110 to 220 volts by means of transformers.

The energy of an alternating current is expressed by the following formula:

$$W = E \times I \times P.$$

W = energy in watts.

E = the voltmeter reading, or voltage.

I = the ammeter reading, or current.

P = power factor.

The above formula is for single-phase alternators. For two-phase, the formula is:

$$W = 2 \times E \times I \times P.$$

For three-phase alternators, the formula is:

$$W = 1.73 \times E \times I \times P.$$

The power factor depends upon the character of the load and the circuit, and is the ratio of the true power to the apparent power, the true power being obtained by the watt-meter reading, and the apparent power by the product of the voltmeter and ammeter readings. It is unity for incandescent lamp circuits, about 0.85 for lights and motors, and 0.8 or less for motors only.

In the case of P = 1, with direct current,

$$W = E \times I.$$

W is in voltamperes.

One thousand voltamperes = one kilovolt ampere (K. V. A.).

Alternating currents are well adapted for long transmissions, but in short transmissions there is no material gain over direct current. For a single plant, if not spread over too great an area, the direct current is preferable. For incandescent lighting it makes practically no difference what kind of current is used, but for power it does, as a direct current motor cannot be used on an alternating circuit or vice versa.

Alternating dynamos and motors require no commutators, but simply collector rings, and sparking is thus entirely avoided. There are two distinct kinds of alternating motors, the *induction* and the *synchronous* motor, and each of these is again divided into *single-phase* and *poly-phase* motors. Single-phase motors require special starting devices, as they will not move from a dead rest when thrown into circuit. There are, however, some self-starting single-phase motors, but they do not give general satisfaction. Two and three phase motors are self-starting. A two-phase system gives the best regulation at the generator, but the three-phase system is more economical in line wires.

The difference between a two-phase and a three-phase motor is simply that the windings of the former are in two groups and of the latter in three groups.

The *synchronous* motor must be excited from some direct

current source, while its armature takes current from the alternating current line. The *induction* motor has its field supplied from the alternating current circuit, and the armature has no connection with any current source, but has its current induced by the fields. The two motors are constructed quite differently from each other. The synchronous motor is constructed almost exactly like the alternator, and if the motor has the same number of poles as the alternator, it will run at the same speed as the alternator, regardless of the load or voltage. The induction motor has its field magnet built up of thin discs instead of solid cores. The revolving part, which is called the armature or the rotor, is also made up of discs, and the windings on it are not laid in slots, but in holes around the edge. The speed of an induction motor can be calculated by multiplying twice the frequency by 60, and dividing the product by the number of poles per phase. This gives the speed per minute.

Operation of Induction Motors.

Troubles on induction motors are often quite perplexing. The output of this motor is limited at any given voltage and is apt to stop when the load becomes too great, and the only way to remedy this is to remove the extra load. As a rule induction motors can carry overloads as high as 50% to 75% above normal much more satisfactorily than direct current motors. It should be remembered, however, that the greater the allowable maximum output, the poorer will be also the power factor and efficiency at normal load.

The journals of the motor need special attention, as it sometimes happens that the armature, due to wear in the bearings, rubs against the field and thus causes a shut-down. Air-gaps should therefore be well examined.

Faults in winding are not an uncommon cause of trouble in starting the motor. Often the compensator is to be blamed by wrong coil connection inside.

There should be about $^1/_{16}$ end play provided for the rotor shaft, and the frequent trouble of oil leaking from the bearings should be effectively stopped.

In buying induction motors it is of interest to know the following:

Efficiency at various loads.

Power factor at various loads.
Apparent efficiency at various loads.
Maximum output at normal voltage.
Current and torque at starting.
The "running light" current.
Drop in speed at various loads.
Torque from rest to synchronism.
Temperatures of the different parts.

Converters and Transformers.

A rotary converter is a machine built in appearance like a direct current multi-polar dynamo or motor, provided with collector rings, and at the same time with a commutator and brushes. It can be used to convert alternating current into direct current or vice versa. It is usually employed to convert alternating current into direct current. In this case it receives the alternating current like a synchronous motor and delivers direct current from the commutator brushes. In the reverse case it runs like a direct current motor and delivers an alternating current from the collector rings. A rotary converter of this kind can be used for various purposes. It may be driven by belt or engine to give out either direct or alternating currents, or both, or it may be driven as a motor and supplied with either current. It may thus be seen that a rotary converter can be used almost anywhere. Almost all substations for electric railroads in America are equipped with rotary converters.

Motor Generators.

For transforming direct current from a higher to a lower voltage a so-called *dynamotor,* sometimes called *motor-generator,* is employed. This consists of a motor and a dynamo mounted on the same base and shaft. The motor receives the high voltage current, and the dynamo gives out a low voltage current. These machines are regular direct current step-down transformers. Direct current step-up transformers, usually called *boosters,* are machines of the same type, with the exception that the dynamo is wound to deliver a higher voltage than the motor receives. Such booster machines are often used for the charging of storage batteries.

Transformers.

For transforming an alternating current either up or down in voltage no rotary transformer is required, a simple stationary

transformer being all that is necessary. This stationary transformer consists of an iron core of thin sheets wound with two sets of coils, namely, the primary and the secondary winding. The high E. M. F. enters the primary coil and creates lines of force in the core which are interlinked by the secondary wires, and consequently an E. M. F. is induced in this winding. Since the E. M. F. is proportional to the product of number of coils, flux and frequency, and since the flux and frequency are the same for both windings, the only difference in E. M. F. must be between the number of turns in the primary and secondary windings. For practical purposes the ratio of turns between the two coils determines the ratio of volts between the primary and secondary coils. The most common ratios are 10 or 20 to 1, and the most common voltages generated with alternators are 1,100 or 2,200 volts, which, if transformed to 110 volts, require the above mentioned ratios between primary and secondary windings. The rise in temperature of a transformer depends entirely on the resistance of the winding and the effective area of the radiating surface. A common rule for the area of wire is to allow about 1,500 c. m. (circular mils) per ampere, and the rise in temperature should not exceed 90° F. If the secondary winding has more turns than the primary winding, the volts are increased, and the device is called *step-up transformer*. Strictly speaking, there is essentially no difference between a step-down, step-up and an ordinary transformer. The latter is a step-down transformer, but the step-up and step-down are built only for high voltage at the secondary end, and extremely high voltage at the primary end, and vice versa.

If the number of primary windings equals the secondary, then the primary voltage equals the secondary. Such transformers are in use for switchboard apparatus, for voltmeters and amperemeters, phase indicators, etc. They are necessarily very small, as they do not deliver any energy. As regards the construction of transformers, there are two distinct types in use:

The *core type:* The windings surround the iron core.

The *enclosed type:* The iron core surrounds the windings.

The *alternating current transformer* is especially well adapted for power and light distribution, as its secondary voltage does not vary with the load in the secondary circuit; it remains constant

as long as the primary voltage is kept constant, and in this way is in itself a self-regulating device.

The efficiency of the transformer depends upon the load, and varies from 90% at light load to as high as 98% at full load.

The transformers in use for power and light transmission and distribution work almost exclusively in parallel, with one exception, for series arc lamp systems, where the so-called *constant current transformer* regulates the voltage whenever lamps are inserted or put out of service.

Electric Lighting.

When an electric current flows through a wire, one part of the electrical energy is turned into heat as a consequence of the

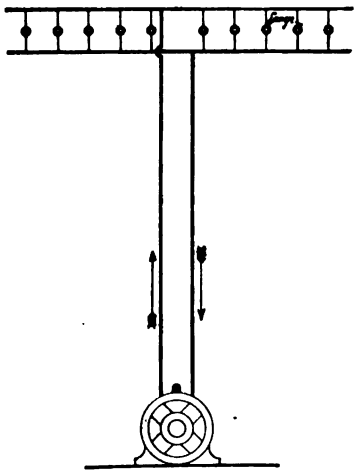


Figure 239.
Lamps Connected in Parallel.

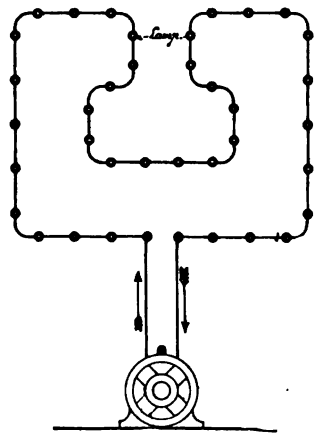


Figure 240.
Lamps Connected in Series.

resistance to the flow of the current. If the wire has a large cross-sectional area and is a good conductor of electricity, the amount of heat developed is too small to be recognized by the touch. On the other hand, if a fine wire or a poor conductor of electricity is used, and an electric current of sufficient potential sent through, it will be heated to such an extent as to become incandescent and produce light.

Incandescent Electric Lamps.

The current passes through a filament of carbonized material enclosed in a glass bulb, from which the air has been exhausted.

The carbon filament is heated to a state of incandescence, about 2,700° F., but as there is no air in the bulb, the carbon only glows and does not burn, as it would do if heated in the air. The carbon loop is joined to two platinum wires which are secured air-tight into the glass. Platinum is used because its coefficient of expansion and that of glass are very nearly the same, so that the heating and cooling of the wire will therefore not cause the air to leak into the bulb. The two platinum wires are connected to a screw piece and a brass contact piece.

. Of the electrical energy expended in the lamp 95% is wasted as heat, and only 5% is represented by the light emitted.

The unit of light is the *candle-power* or the equivalent of the light emitted by a spermaceti candle ⅞ of an inch in diameter, which is so proportioned as to burn 120 grains per hour. The light given off by a lamp is usually tested with a *Bunsen photometer.* Lights are tested as to their mean spherical candle-power and also their mean horizontal candle-power. Of the Edison type of lamp the 16 candle-power size is in most general use, and such a lamp requires on an average from 55 to 60 watts. The average life of such lamps is about 600 hours. The heat developed in an incandescent lamp is approximately 200 B. T. U. per hour. Incandescent lamps, if operated on a constant potential circuit, are usually connected in parallel (see Figure 238), but on a constant current circuit they are either connected in series or in multiple (see Figure 239). The latter case is very seldom encountered, as the burning out of one lamp interrupts the current, and therefore the rest of the lamps in series are out of service.

Great strides have been made in the last few years in improving lamps to better efficiency. They are classed under different names, such as *Osmium, Tantalum, Mazda, Tungsten* lamps, etc. Even the carbon lamps have been superseded by the slightly more economical *Gem* type metallized filament lamps. The current consumption of the Tungsten lamps is practically one-half that of the carbon lamp, and great savings have been effected in cases where the lighting load is a considerable item. Some power plants have gone on record in reducing their lighting load to such an extent by the adoption of more economical lamps as to cause the shut-down of a complete generator unit.

The use of the higher priced but also more economical

Tungsten lamps is justified even in isolated plants manufacturing their own electricity, and is not at all restricted to establishments supplied by outside central station service.

Voltage Regulation.—Variations of voltage have always been a cause for the shortening of the life of the lamps and for the fluctuation of candle-power of the lamp affected.

One per cent of voltage variation will produce a candle-power fluctuation of 5.6% with the Carbon lamp, 4.8% with the Gem lamp and 3.7% with the Tungsten.

Lamps are now usually rated at three voltages. Operating the lamps at the highest voltage on the label gives highest efficiency and shortest life, at the lowest voltage, which is 4 volts less than top voltage, gives lowest efficiency and longest life.

A 60-watt Carbon lamp will produce at medium efficiency 18.3 candle-power. Under like efficiency and wattage the Gem lamp will give 21.9 C. P.,the Tantalum 30 C. P. and the Tungsten 47.3 C. P. The watts per candle at medium efficiency are for:

Carbon	3.18 watts
Gem	2.65 watts
Tantalum	1.87 watts
Tungsten	1.23 watts

which demonstrates the large saving obtainable.

Tungsten lamps are made in all sizes up to 500 watts, and these are successful competitors to arc lamps, as they use considerably less current, besides doing away with the labor attendance that is required for the trimming of arc lamps.

Good reflectors are a good investment, as they help to diffuse and distribute the light.

Arc Lights.

If a strong electric current is sent through a wire, the wire cut, the ends of the wire again brought together and then separated a short distance, a small but very intense light will be produced between the separated ends. The heat produced by the resistance of the contact surfaces of the ends of the wire volatilizes a part of the material of the wire, and this gas serves as a very high resistance across the air-gap.

The interruption of a circuit in this way is what is known as a *voltaic arc,* or simply arc. The heat produced by such an arc is so great that a metal such as platinum melts like tallow. For this

reason rods of specially prepared carbon (made from gas coke) are employed, they being less fusible than any other substance. The volatilization of the carbon takes place chiefly at the end of the positive carbon, which, therefore, gradually wastes away, forming a cup-shaped hollow, called the *crater,* at the end of the rod, whereas the negative carbon takes the form of a cone. For direct current the positive carbon is made twice as thick as the negative. The light approaches the sunlight in strength and whiteness, and the temperature in the crater is estimated at between 6,300° F. and 7,500° F.

When, on account of the wearing away of the carbons, the distance between the two rods becomes too great for the electric current to pass over, the light goes out. In the open electric arc lamp, self-acting regulators maintain a uniform distance between the poles. The light of an ordinary street lamp has a light capacity varying from 1,200 to 2,000 candle-power and sometimes is rated even higher. Such lamp circuits are usually of the constant current type, in which the lamps are all connected in series, the voltage for each lamp varying between 45 and 50 volts for direct current, and 30 to 35 volts for alternating current across the arc, and the current strength varies between 6 and 10 amperes. With alternating current there is not a continuous flame, as the arc is lighted and extinguished with every reversal of the current. However, the flicker is not noticeable when the number of alternations is about 100 per second. A 30% greater current is required for the alternating arc than for the direct current arc in order to produce sufficient carbon vapor to maintain the path at the reversals of the current.

For the purpose of prolonging the life of the carbons and to obtain a more diffused light, the carbons are enclosed hermetically in a glass globe, whereby the oxygen is prevented from burning them. Such lamps burn about 10 to 15 times as long as those with open air arc, but they require a voltage about twice as high.

There are a great many devices invented to feed the carbon at a rate equal to the burning off. They vary greatly with the source, if direct current or alternating current, or system, if series or parallel, and with the difference in design of the various manufacturers. Flaming arc lamps are powerful light units, requiring

trimming oftener than the common arc lamp. Their carbons last about 16 hours.

The *Nernst lamp* takes a place between the arc lamp and the carbon incandescent lamp and is far more economical than the latter. The 16-candle power Nernst lamp takes only 30 watts in a 100-volt circuit. They are made for direct current and alternating current, but give better satisfaction with the latter. It is necessary to heat the light-emanating medium a short time before the lamp gives light, which is done automatically by the current flowing first through a platin-solenoid wound around the medium consisting of a rod or a metal oxide, i. e., magnesium.

Another unique lamp, using mercury vapor as a medium, is the powerful and economical *Cooper-Hewitt lamp*. Its strong green-bluish light makes it adaptable for shop and factory use.

Electrical Calculations.

The *heat developed by a lamp* may be calculated, if the number of watts consumed per hour by the lamp is known.

Example.—Assuming that a 16 candle-power lamp requires 56 watts, how many B. T. U. are developed per hour?

Solution.—1 joule = 1 watt-second.

1 watt-hour = 3,600 joules.

1 joule = 0.7373 foot-pound, hence 56 watts = $3{,}600 \times 56 \times 0.7373 = 148{,}639.68$ foot-pounds or $\frac{148{,}639.68}{778}$ = 191.05 B. T. U. per hour.

This calculation may also be accomplished as follows: If the lamp burns for one hour, it will consume

$$\frac{56}{1{,}000} \times 1 \text{ hour} = 0.056 \text{ K. W. hour.}$$

From conversion tables, it is found that 1 K. W. hour = 3,412 B. T. U.

Therefore the heat produced per hour = $0.056 \times 3{,}412$ = 191.07 B. T. U.

The proper *size of dynamo* for a given number of lamps and motors under assumed simultaneous operation may be calculated as follows:

Example.—Let the efficiency of the dynamo be 92%, and of the motors 85%, the drop in the motor circuit 5% volts and in the lamp circuit 2½%. The dynamo should be capable of

furnishing direct current of 110 volts for 600 16-C. P. lamps, each consuming 56 watts, 12 arc lamps, each consuming 550 watts, and two 25, three 10 and four 5 H. P. motors?

Solution.—$(600 \times 56) + (12 \times 550) = 40{,}200$ watts, which must be furnished to the lamps; the loss in the circuit, however, is 2½%, hence the circuit must receive $\frac{40{,}200}{0.975} = 41{,}230$ watts for lighting purposes.

The motor circuit requires: $(2 \times 25) + (3 \times 10) + (4 \times 5) = 100$ H. P., or $100 \times 746 = 74{,}600$ watts, if there were no losses in the motors and in the motor circuit; but since the efficiency of the motors is 85%, then the watts required by the motors are $\frac{74{,}600}{0.85} = 87{,}765$ watts. To this must be added the loss in the motor circuits, which is 5%, hence, $\frac{87{,}765}{0.95} = 92{,}384$ watts, are required for the motor circuits. The dynamo must therefore furnish $41{,}230 + 92{,}384 = 133{,}614$ watts, and since the efficiency of the dynamo is 92%, it must have a capacity of $\frac{133{,}614}{0.92} = 145{,}233$ watts. The nearest size machine obtainable in the competitive market would be 150,000 watts, or 150 K. W.

The proper size engine for this dynamo, assuming 10% friction of engine and dynamo, would be,

$\frac{150{,}000}{746 \times 0.90} = 223.5$, or in round numbers 225 H. P. indicated.

Dimensions of Feeder Cables from Switchboard to Distributing Panel.

In the following, a short description and calculations will be offered for the correct design of an assumed system of light and power wiring supplied by the above dynamo.

It should be stated that for reasons of economy, simplicity and safety it is necessary to arrange for so-called centers of distribution, which must be centrally located in relation to the lamp or motor circuits to be served. These centers of distribution, also known as *light distribution panels* or *power distribution panels*, are connected to the switchboard direct by *feeders*.

The distributing panels consist of the feeder bus bars, and leading off from them are a number of branches, properly fused, connected to the many smaller branch circuits for motors or lights.

It is recommended to separate the light circuits from the motor circuits, if a constant voltage is required on the light. Such panels must be put up in boxes of metal or fireproof material.

The feeder cables must be calculated of sufficient size in order to carry the load of all motors and lights without undue heating.

The proper size wire for the lamp and motor feeder cables for the example previously given may be calculated as follows: Assume there are three incandescent lamp circuits, one arc circuit and three motor circuits, each supplying the following quantities at different lengths of each circuit. By the length of a circuit is meant the distance from the switchboard to the center of lamp or power distribution.

The arc lamp circuit:

(a) supplying 12 lamps is 150 feet.

The three incandescent lamp circuits are supplying:

(b) 175 lamps at a distance of 350 feet.

(c) 200 lamps at a distance of 300 feet.

(d) 225 lamps at a distance of 160 feet.

The three motor circuits are supplying:

(e) Two 25 H. P. motors at a distance of 60 feet.

(f) Three 10 H. P. motors at a distance of 85 feet.

(g) Four 5 H. P. motors at a distance of 125 feet.

In order to calculate the proper size wire in circular mils three factors must be known, namely:

Length of circuit in feet = distance $\times$ 2.

Number of amperes, I, wire should supply.

Number of volts, E, lost between switchboard and center of distribution.

Then if L = length of circuit in feet, I = amperes required, E = drop (volts lost in circuit), c. m. = circular mils of wire required, the formula is:

$$\text{c. m.} = \frac{L \times I \times 10.8}{E}$$

$$E = \frac{L \times I \times 10.8}{\text{c. m.}}$$

$$I = \frac{\text{c. m.} \times E}{L \times 10.8}$$

The constant 10.8 is the number of circular mils in a copper wire which would have a resistance of one ohm for one foot of

length at ordinary temperatures. A circular mil is the area of a circle whose diameter is one-thousandth part of one inch.

In order to solve the above example, it becomes necessary to calculate the number of amperes each circuit should supply, and also the number of permissible volts drop in each circuit. The following are the amperes for the four lamp and three power circuits respectively, considering a drop of voltage of 2½%, equal to 97.5% efficiency.

$$\text{Arc lamps} = \frac{12 \times 550}{0.975 \times 110} = 61.5 \text{ amperes for (a).}$$

$$\text{Incandescent lamps} = \frac{175 \times 56}{0.975 \times 110} = 91.366 \text{ amperes for (b).}$$

$$\text{Incandescent lamps} = \frac{200 \times 56}{0.975 \times 110} = 104.43 \text{ amperes for (c).}$$

$$\text{Incandescent lamps} = \frac{225 \times 56}{0.975 \times 110} = 117.48 \text{ amperes for (d).}$$

$$\text{Motors} = \frac{2 \times 25 \times 746}{0.85 \times 0.95 \times 110} = 420 \text{ amperes for (e).}$$

$$\text{Motors} = \frac{3 \times 10 \times 746}{0.85 \times 0.95 \times 110} = 252 \text{ amperes for (f).}$$

$$\text{Motors} = \frac{4 \times 5 \times 746}{0.85 \times 0.95 \times 110} = 168 \text{ amperes for (g).}$$

$$\text{The drop for the lamp circuits} = \frac{2.5 \times 110}{100} = 2.75 \text{ volts.}$$

$$\text{The drop for the power circuits} = \frac{5 \times 110}{100} = 5.5 \text{ volts.}$$

We have now the values of I, E and L, hence the c. m. for each feeder-circuit are:

$$\text{Arc lamp circuit (a)} \frac{150 \times 2 \times 61.5 \times 10.8}{2.75} = 72{,}458 \text{ c. m.}$$

or $\frac{\sqrt{72{,}458}}{1{,}000} = 0.270$ of an inch in diameter.

The nearest size upward of wire that should be used would be a No. 1 B. & S. gauge wire, having an area of 83,690 c. m. The sizes of these wires can be ascertained from any Underwriters' table of approved and standard size wires.

$$\text{Incandescent circuit (b)} \frac{350 \times 2 \times 91.366 \times 10.8}{2.75} = 251{,}173 \text{ c. m.,}$$

equal to a 300,000 c. m. cable.

Diameters of Copper and Steel Wire by Different Gauges, and Sheet-Metal Gauge.

Decimals of an Inch.

Number	Birmingham or Stubbs.	Roebling's, and Washburn and Moen's Gauge.	Brown and Sharpe.	English Legal Standard.	U. S. Standard Gauge for Sheet and Plate, Iron and Steel, 1893.	Stubb's Steel Wire Gauge.
0000000	...	.49	.....	.500	.500	...
000000	...	.46	.....	.464	.469	...
00000	...	.43	.....	.432	.438	...
0000	.454	.393	.46000	.400	.406	...
000	.425	.362	.40964	.372	.375	...
00	.380	.331	.36480	.348	.344	...
0	.340	.307	.32486	.324	.313	...
1	.300	.283	.28930	.300	.281	.227
2	.284	.263	.25763	.276	.266	.219
3	.259	.244	.22942	.252	.250	.212
4	.238	.225	.20431	.232	.234	.207
5	.220	.207	.18194	.212	.219	.204
6	.203	.192	.16202	.192	.203	.201
7	.180	.177	.14428	.176	.188	.199
8	.165	.162	.12849	.160	.172	.197
9	.148	.148	.11443	.144	.156	.194
10	.134	.135	.10189	.128	.141	.191
11	.120	.12	.09074	.116	.125	.188
12	.109	.105	.08081	.104	.109	.185
13	.095	.092	.07196	.092	.094	.182
14	.083	.08	.06408	.080	.078	.180
15	.072	.072	.05707	.072	.070	.178
16	.065	.063	.05082	.064	.0625	.175
17	.058	.054	.04526	.056	.0563	.172
18	.049	.047	.04030	.048	.050	.168
19	.042	.041	.03589	.040	.0438	.164
20	.035	.035	.03196	.036	.0375	.161
21	.032	.032	.02846	.032	.0344	.157
22	.028	.028	.02535	.028	.0313	.155
23	.025	.025	.02257	.024	.0281	.153
24	.022	.023	.02010	.022	.0250	.151
25	.020	.02	.01790	.020	.0219	.148
26	.018	.018	.01594	.018	.0188	.146
27	.016	.017	.01419	.0164	.0172	.143
28	.014	.016	.01264	.0148	.0156	.139
29	.013	.015	.01126	.0136	.0141	.134
30	.012	.014	.01002	.0124	.0125	.127
31	.010	.013	.00893	.0116	.0109	.120
32	.009	.013	.00795	.0108	.0101	.115
33	.008	.011	.00708	.010	.0094	.112
34	.007	.01	.00630	.0092	.0086	.110
35	.005	.0095	.00561	.0084	.0078	.108
36	.004	.009	.00500	.0076	.007	.106
37	...	.0085	.00445	.0068	.0066	.103
38	...	.008	.00396	.006	.0063	.101
39	...	.0075	.00353	.0052	....	.099
40	...	.007	.00314	.0048	....	.097
41	...	...	.....	.0044	....	.095
42	...	...	.....	.004	....	.092
43	...	...	.....	.0036	....	.088
44	...	...	.....	.0032	....	.085
45	...	...	.....	.0028	....	.081
46	...	...	.....	.0024	....	.079
47	...	...	.....	.002	....	.077
48	...	...	.....	.0016	....	.075
49	...	...	.....	.0012	....	.072
50	...	...	.....	.001	....	.069

Incandescent circuit (c) $\frac{300 \times 2 \times 104.43 \times 10.8}{2.75} = 246{,}090$ c.m.,

equal to a 300,000 c. m. cable.

Incandescent circuit (d) $\frac{160 \times 2 \times 117.48 \times 10.8}{2.75} = 147{,}640$ c. m.,

equal to a No. 000 cable.

Motor circuit (e) $\frac{60 \times 2 \times 420 \times 10.8}{5.5} = 98{,}967$ c. m.,

equal to a No. 0 B. & S. gauge cable.

Motor-circuit (f) $\frac{85 \times 2 \times 252 \times 10.8}{5.5} = 84{,}122$ c. m.,

equal to a No. 1 B. & S. gauge cable.

Motor circuit (g) $\frac{125 \times 2 \times 168 \times 10.8}{5.5} = 82{,}473$ c. m.,

equal to a No. 1 B. & S. gauge wire.

The above sizes would represent the sizes of the feeder cables which are run from the switchboard feeder switches to the respective distributing panels.

For the wiring of individual branch circuits from the distributing centers, for lights and motors, the rules of the Board of Underwriters must be followed, a short abstract being here presented.

Motor leads are required to be heavy enough to stand 25% overload without undue heating.

Wires smaller than No. 14 B. & S. gauge must not be used for lighting circuits.

For the specific design of three-wire and four-wire D. C. systems, as also alternating wiring systems, other works should be consulted, as this is beyond the scope of this book.

Storage Batteries.

Storage batteries have their own field of adaptation, and in certain cases are almost indispensable. Their installation is often conducive to operating a plant at great economy, but in every instance all points must be considered very carefully.

Their advantages are remarkable in a plant for equalizing the load; that is, where the generator has to overcome certain excessive peak loads during short periods, while during the greater part a lighter load is carried.

In such cases, during the charging period, they will allow the generator to run at maximum economy, and during the discharging period, supply the surplus of power needed to overcome the peak load.

They are furthermore of great service, when a smaller load can be carried during a part of the day or night, permitting the generators and boilers to be shut down completely. This manner of operation is usually economical, if a saving in labor can be effected at the same time. It is seldom economical, however, in cases where the boilers must be kept going and the usual operating force be in attendance for other purposes anyway.

Storage batteries are of advantage to tide over sudden rushes of current, such as elevator service and the like, and as a convenient auxiliary in cases of emergency causing shut-down of the generators for short periods.

For the propelling of automobiles and motor boats they have found quite a resourceful field of application.

In estimating the economy sought for, every angle of their advantages and disadvantages, as well as cost of installation, maintenance and repair, building floor space, labor, efficiency, interest and depreciation should be investigated.

The electric storage battery, also called *accumulator,* consists of a number of cells combined in a proper manner, of which each cell forms a separate unit.

They consist principally of a container, which, in the case of the sulphuric acid batteries, is lined with sheet lead; the plates and the electrolyte.

The plates are positive and negative, placed in the solution alternately with insulating material between to keep them from touching. All positive plates are connected together, as are also all negative plates.

The positive plates are usually made of lead with a coating of peroxide of lead, while the negative plates are of pure lead. As an electrolyte a diluted sulphuric acid bath is used. As soon as electric current is applied to the connected plates a chemical action takes place within the battery, the current entering at the positive pole passes through the acid to the negative. This causes the decomposition of the sulphate of lead, depositing at the same time the oxygen from the negative to the positive. This process

is called *charging the battery*. In *discharging the battery* the reverse takes place, as sulphate of lead is being formed, which simultaneously reduces the specific gravity or strength of the electrolyte. The strength of the solution is tested by means of a hydrometer. Only pure sulphuric acid, free from impurities, should be used.

The charging current should always be applied slowly and extending over a long period of hours.

The *Edison battery* is now being introduced, as it has in some points the advantage over the acid battery in not emitting acid fumes on charge. The electrolyte used is a 21% solution of potash in distilled water with a small per cent of lithia.

The efficiency of a battery varies a great deal according to make and as to manner of operation. The percentage ratio between output and input of current ranges from 50% to 85%.

Load Factors.

The actual output of a plant, generator or motor is always in a certain relation to its full capacity rating. The ratio thus established is called the load factor.

Load factors may be established for any period of time, as, for instance, yearly, monthly, weekly, daily or even hourly, but in each case should be clearly defined and designated.

The load factor plays an important part in the economy of any plant. The higher the ratio of actual output to rated capacity, the higher the load factor and also the better the economy. Most engines, turbines and other prime movers, as well as generators and motors, are designed to give highest efficiency at full load, and from this standpoint alone, besides the necessarily smaller fixed charges due to full load condition, the great economy obtainable from high load factors is obvious.

The integrating wattmeter furnishes the means of establishing the load factor of a plant most accurately, as:

$$\text{Yearly load factor} = \frac{\text{K. W. hours per year.}}{\text{rated capacity in K. W.} \times 8{,}760}$$

8,760 stands for 365 days of 24 hours each, or 8,760 hours per year.

$$\text{Daily load factor} = \frac{\text{K. W. hours per day.}}{\text{rated capacity in K. W.} \times 24}$$

The above load factors apply to plants which are in continual operation, day and night, throughout the year.

In many cases it is desired to know the load factors of plants which operate only a certain number of hours per day and then shut down.

This so-called *curve load factor* is therefore the ratio of the actual output per year to the rated capacity (full load output), depending, however, on the actual number of hours during which the plant is operating; and the formula would be:

$$\text{Curve load factor} = \frac{\text{K. W. hours per year}}{\text{rated capacity} \times \text{actual hours of operation.}}$$

It is evident that labor and the fuel consumption, oil and other supplies vary directly with the number of hours during which the plant is in actual operation, while on the other hand, the fixed charges, such as salaries for chief engineer, possibly for his assistant, and charges for depreciation, interest, taxes and insurance, go on just the same.

The greater the load factor, therefore, the less the fixed charges will become, as they are thus distributed over a greater number of hours, and it is evident that the cost per kilowatt hour becomes less also. A load factor of 100% would, therefore, mean the highest attainable economy, as the machines would run every hour of the year up to their full rating.

To illustrate, if in one given case of a 150 K. W. generator, this machine would be operated at full load for 8,760 hours per year, the fixed charges per year being $1,971.00, then the fixed charges per K. W. hour under these conditions would be

$$\frac{\$1{,}971.00}{150 \times 8{,}760} = \$0.0015.$$

If, however, the load factor is less than 100%, as, for instance, 60%, then the fixed charges per K. W. hour increase accordingly, or

$$\frac{\$1{,}971.10}{0.60 \times (150 \times 8{,}760)} = \$0.0025.$$

The yearly load factor in this case could be calculated, provided the number of K. W. hours per year are known to be 788,400 K. W. hours, as follows:

$$\frac{788{,}400}{150 \times 8{,}760} = 0.60 = 60\% \text{ load factor.}$$

If the records show an expenditure of $4,415.04 for operating expense per year, the operating expense per K. W. hour is found by

$$\frac{\$4,415.04}{788,400} = \$0.0056 \text{ per K. W. hour.}$$

The actual total cost of producing one K. W. hour is therefore found by adding the two charges together, or:

Operating expense	$0.0056
Fixed charges at 60% load factor..	0.0025
Total cost per K. W. hour.....	$0.0081

To reiterate, if the load factor would have been higher, it is plain that the operating expense as well as the fixed charges, and therefore the total cost per K. W. hour, must become less.

Cental Stations versus Isolated Stations.

This question is being ventilated every day, and the engineer should acquaint himself with the facts and figures obtainable in the plant under his charge, and by such strict methods of keeping records he will be able to show figures of results.

The central station can seldom or never compete in plants where a careful system of exhaust steam utilization has been worked out, as the efficiency of such a plant naturally must be very much higher than that of a central station which carries off most of the heat contained in the steam in their condensing water. Furthermore, where the operating force cannot be reduced, as in the case of running ice machines and the like, which require attention, the chances are slim for the central station, as the engineer attending the ice machines, etc., may easily take care of the engine driving the same and the generator besides without extra effort or expense.

In cases, however, which permit the elimination of the boiler room and its operating force, and where heat in the form of steam or exhaust steam is not essential, the field for the central station is unquestionable.

Heating and Ventilation

Generally speaking, there are only two methods of heating buildings, viz., by means of steam or by hot water, but on the former are based a great number of different systems. In the first place, we can distinguish between direct and indirect steam heating; either one of the two may be operated by the so-called one-pipe or by the two-pipe system; the steam employed may be at a high or at a low pressure; the regulation of the steam may be automatic or it may be done by hand. The following paragraphs will give a short description of the systems in general use.

Indirect Steam Heating.

As the name implies, the indirect steam heating method does not use the steam for heating radiators placed directly in the rooms. Heating coils or radiators in a sufficiently large number for heating all the rooms are placed in one room, usually in the basement, and the steam from the boiler or exhaust steam is circulated through them.

Fresh air is drawn through the heating chamber by a fan. This is done at a low velocity, in order to enable sufficient heat to be absorbed from the coils. The fan delivers the warm air into air ducts leading to the various rooms.

In this manner the rooms are ventilated at the same time as they are heated, in fact one is impossible without the other.

Automatic Regulation of Temperature.—In order to maintain a constant temperature in the rooms, even if the outside temperature or the temperature in the heating chamber should vary considerably, a mixing damper is placed between the fan and heating coils controlled by a thermostat connected to an automatic diaphragm valve. The admission of air is thereby automatically regulated according to the temperature, less air

being admitted in case the outdoor temperature or that of the heating chamber is low, and more if the reverse is the case.

Dimensions of Air-Ducts.—The main air-duct must be of such dimensions that with proper heating and ventilation of all the rooms, the velocity of the air does not exceed 600 feet per minute. A greater velocity than this would cause annoyance to occupants of the room.

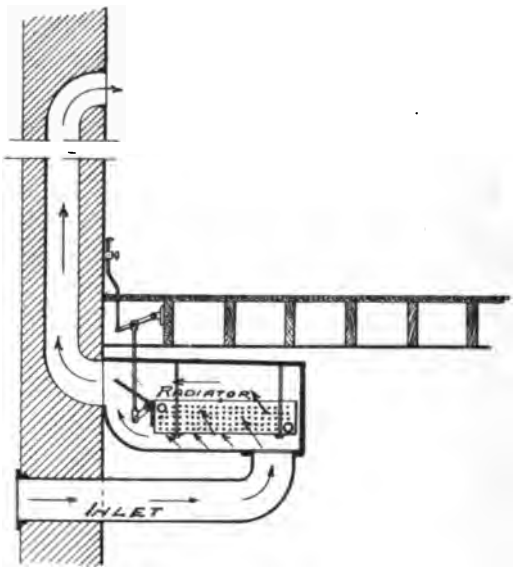


Figure 241. Indirect Steam Heating System.

The area of an air-duct can be calculated, if the quantity of air to be delivered in a given time is known. The formula is as follows:

$$A = \frac{Q}{v}$$

A = area of air-duct in square feet.
Q = cubic feet of air delivered per minute.
v = velocity of air in feet per minute.

Example.—What must be the area in square feet of an air-duct which is to deliver a sufficient amount of air for a school of 1,000 pupils, if each pupil requires 30 cubic feet of

air per minute, and if the velocity of the air is to be 600 feet per minute?

Solution.—The total quantity of air required per minute is $1{,}000 \times 30$ cubic feet, consequently:

$$A = \frac{1{,}000 \times 30}{600} = 50 \text{ square feet.}$$

If a round air duct is used, the diameter of the pipe would, therefore, have to be:

$$d = \sqrt{\frac{50}{0.7854}} = 8 \text{ feet.}$$

Amount of Air Required for Ventilation.—The quantity of air required is not by any means invariable; it depends on the purpose for which the building is intended, the sanitary conditions, the number and age of the occupants of the building, and other factors. Average figures for the amount of air required per person are given in the following table:

Cubic Feet of Air Required per Minute per Head for Ventilation.

Hospitals, ordinary	45 cubic feet
Hospitals for wounded	60 cubic feet
Hospitals for contagious diseases	90 cubic feet
Schools for infants	15 cubic feet
Schools for older children	30 cubic feet
Theaters	36 cubic feet
Assembly halls	35 cubic feet
Prisons	35 cubic feet
Workshops	35 cubic feet
Workshops, poor sanitary conditions	60 cubic feet
Barracks	35 cubic feet
Stables	120 cubic feet

Calculation of Radiating Surface for Indirect System.—When calculating the necessary radiating surface for heating a building according to the indirect method, a great many different factors must be taken into consideration. In the first place, the loss of heat through walls, floor and ceiling must be known, and this, in its turn, requires the knowledge of various factors. It may be calculated by using the following formula:

$$L = S \times k \times (t - t_1).$$

L = loss of heat in B. T. U. per hour.
S = number of square feet of surface (walls, floor, ceiling).
t = inside temperature in degrees F.
t_1 = outside temperature in degrees F.
k = B. T. U. transmitted by one square foot of surface per hour per degree difference between outside and inside temperatures.

The factor k varies with the material used in the construction of the building, and for brick walls also varies with the thickness. The following tables, which give the different values of k, have been calculated by A. R. Wolff.

B. T. U. Transmitted by One Square Foot of Brick Wall.

Thickness of Wall in Inches	4	8	12	16	20	24	28	32	36	40
k	0.68	0.46	0.32	0.26	0.23	0.20	0.174	0.15	0.129	0.115

B. T. U. Transmitted by One Square Foot of Various Materials Per Hour, Per Degree F. Difference Between Outside and Inside Temperatures.

Material	*k*
Wooden floor	0.083 B.T.U.
Fireproof floor	0.124 B.T.U.
Wooden ceiling	0.104 B.T.U.
Fireproof ceiling	0.145 B.T.U.
Single window	1.030 B.T.U.
Single skylight	1.118 B.T.U.
Double window	0.518 B.T.U.
Double skylight	0.621 B.T.U.
Door	0.414 B.T.U.

Where the surface is exposed to the north or to continuous winds, the value of k should be increased 10%. If the building is to be heated in day-time only, the value of k must be increased 30%, which means that the radiating surface has to be 30% larger.

For calculating the required radiating surface, the number of persons occupying the room or rooms, and consequently the quantity of air necessary for ventilation, must be known. The

number of lights burned, and whether gas or electricity is used, must also be taken into consideration.

The radiating power of the radiators must also be known. The latter has been determined by A. R. Wolff as amounting to 400 B.T.U. per hour per square foot for non-painted radiators with steam pressure of from 3 to 5 pounds, and to 250 B.T.U. for bronzed cast iron radiators under the same conditions.

The proper method for calculating the radiating surface is illustrated by the following example.

Example.—A lecture room for 40 people is to be heated by the indirect heating system. The room is 25 feet long, 24 feet wide, and 20 feet high. The north wall is 25×20 feet and has two 14×4 feet windows. The south wall has two windows of the same dimensions. Adjoining the west wall is a corridor which is kept at 70° F. The east wall is exposed to the outside air and has a door 10×4 feet. The basement under the lecture room has a temperature of 40° F. The ceiling is exposed to the outside air, the temperature of which is 0° F. The walls are 20 inches thick, and floor and ceiling are of fireproof construction. A temperature of 70° F. is required in the room. How many square feet of radiating surface are necessary?

Solution.—The surface of walls, floor, ceiling, windows and door, and the loss of heat through them, must first be calculated.

Surface of north wall ((less windows)
.....$25 \times 20 - 2 \times 14 \times 4 = 388$ square feet
Surface of south wall (less windows)
.....$25 \times 20 - 2 \times 14 \times 4 = 388$ square feet
Surface of east wall (less door)
.....$24 \times 20 - 4 \times 10 = 440$ square feet
Surface of floor .....$25 \times 24 = 600$ square feet
Surface of ceiling .....$25 \times 24 = 600$ square feet
Surface of west wall (must be omitted, as there is no loss of heat through it).
Surface of north wall windows...$2 \times 4 \times 14 = 112$ square feet
Surface of south wall windows...$2 \times 4 \times 14 = 112$ square feet
Surface of east wall door.....$4 \times 10 = 40$ square feet

By using the values of k and the formula given above, we find the following heat transmission through walls, etc.

North wall .....$388 \times 70 \times 0.23 + 10\% = 6{,}872$ B.T.U.
North wall windows..$112 \times 70 \times 0.776 + 10\% = 6{,}692$ B.T.U.
South wall .....$388 \times 70 \times 0.23 = 6{,}247$ B.T.U.
South wall windows.....$112 \times 70 \times 0.776 = 6{,}084$ B.T.U
East wall .....$440 \times 70 \times 0.23 = 7{,}084$ B.T. U.
East wall door.....$40 \times 70 \times 0.414 = 1{,}160$ B.T.U.
Floor .....$600 \times (70 - 40) \times 0.124 = 2{,}232$ B.T.U.
Ceiling .....$600 \times 70 \times 0.145 = 6{,}090$ B.T.U.

Total heat transmission.....42,461 B.T.U.

After having thus found the total amount of heat lost by transmission, we have to add to it the quantity of heat required to warm the air for ventilation. There are 40 persons, and each person requires about 35 cubic feet of air per minute. This air must be heated from 0° to 70° F., which takes about 1.4 B.T.U. per cubic foot, and the total number of B. T. U. necessary per hour for the air for ventilation therefore amounts to $40 \times 35 \times 60 \times 1.4$ or 117,600 B.T.U.

However, the 40 occupants of the room give off a considerable quantity of heat, approximately, 400 B.T.U. per person per hour, or 16,000 B.T.U. in all, and this heat must be subtracted from the total amount required.

Assuming the room to be lighted by 8 gas burners, the heat generated by them would amount to $8 \times 3{,}250$ or 26,000 B.T.U. per hour. If 12 electric lights were used instead, 12×190 or 2,280 B.T.U. would be generated. Since, however, the room is used almost exclusively during the day-time, the heat developed by the lights can be omitted from the calculations and, therefore, is not to be subtracted from the heat required.

The total quantity of heat necessary for heating the room is as follows:

Heat of transmission..... 42,461 B.T.U.
Heat of ventilation .....117,600 B.T.U.

160,061 B.T.U.
Less heat given off by occupants..... 16,000 B.T.U.

Total .....144,061 B.T.U.

Since about 400 B.T.U. per hour are given off by one square foot of heating-coil surface, there are $\frac{144,061}{400}$ or 360 square feet of radiating surface necessary.

Empirical Rules for Calculation of Radiating Surface.—From the foregoing it can readily be seen that many factors enter into the calculation of the radiating surface, and since the factors change frequently, exact results are difficult to obtain. Empirical rules, are, therefore, often employed, certain factors having been determined from actual experience, which give the number of cubic feet heated by one square foot of radiating surface in localities serving various purposes. The following table gives these ratios between radiating surfaces and cubic contents for the indirect steam heating system.

Cubic Feet of Space Heated by One Square Foot of Radiating Surface.

Indirect System.

Residences, schoolrooms, offices, etc.....	40	to	50 cubic feet
Assembly halls, stores, factories, etc.....	50	to	70 cubic feet
Churches, large auditoriums, etc.....	100	to	140 cubic feet

For a lecture room 25 feet long, 24 wide and 20 high there would consequently be necessary $\frac{25 \times 24 \times 20}{40}$ or 300 square feet of radiating surface.

Size of Boiler.—The dimensions of the boiler required for the heating system depend not only on the radiating surface but also on the size of the building. For residences, one square foot of boiler surface should be allowed for each 7 square feet of radiating surface, while for larger buildings, a ratio of 1 to 10 is sufficient. Each boiler horse-power will supply from 80 to 120 square feet of radiating surface.

Direct Steam Heating.

In the system of direct steam heating, the steam is circulated through radiators of suitable size, which are distributed in the various rooms. High pressure steam or exhaust steam may be employed, according to conditions. Ventilation of the rooms must be done separately.

In some buildings, the radiators are not placed directly in the room, but are enclosed in a small chamber in the wall, usually under the window; steam is kept on continually and the heat is admitted to the room or shut off by means of a register with which the radiator case can be closed at will. This system, which is known as the *direct indirect* heating system, is, however, being gradually abandoned, since the heat of the steam is not sufficiently utilized.

Calculation of Radiating Surface for Direct Heating System.—A somewhat different method must be followed in calculating the radiating surface necessary for a direct steam heating system than was employed for the indirect system. In

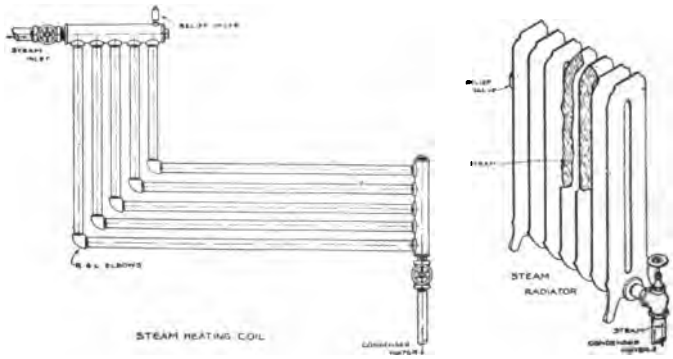


Figure 242. Direct Steam Heating System.

the latter, heating and ventilation are one, while when heating a room directly, the amount of air necessary for ventilation must be assumed for the calculation. Professor Carpenter states that it is safe to assume that the air of the principal living room of a residence is changed twice in one hour, that of halls three times and that of the other rooms once per hour. The same author gives the following formula for calculating the radiating surface for a direct heating system:

$$h = \frac{n}{55} \times c + g + 0.25 \times w.$$

h = number of B.T.U. required per hour, per degree F. difference between outside and inside temperatures.
n = number of times air is changed per hour.
c = cubic feet of space in building.

g = window or other glass surface in square feet.
w = wall surface in square feet.

Example.—How many square feet of radiating surface are required for a lecture room 25 feet long, 24 wide and 20 high, having 4 windows 14 × 4 feet each? The air is renewed three times per hour?

Solution.—According to the data given above:
n = 3.
c = 12,000 cubic feet.
g = 224 square feet.
w = 1,736 square feet.

Hence: $h = \frac{3}{55} \times 12{,}000 + 224 + 0.25 \times 1{,}736 = 1{,}313$ B.T.U.

This means that 1,313 B.T.U. are required per hour per degree F. difference between outside and inside temperatures. If, therefore, the outside temperature is 0° and if the room is to be kept at 70°, then 1,313 must be multiplied by 70 in order to obtain the total number of B.T.U. required. Since one square foot of bronzed radiating surface transmits on the average 280 B.T.U. per hour, the radiating surface necessary amounts to $\frac{1{,}313 \times 70}{280} = 328$ square feet.

If the air in the room were to be renewed only once per hour, the radiating surface would have to be 219 square feet, and if the air is changed every 3 hours, then 183 square feet would be required, as can easily be calculated by means of the above formula.

Empirical Rules.—From the above calculation it can be seen that ventilation plays a very important role in the calculation of radiating surface, and since ventilation is a very uncertain quantity with the direct heating system, empirical rules are generally followed, similar to the ones given for the indirect system. The following table gives the factors generally employed:

Cubic Feet of Space Heated by One Square Foot of Radiating Surface.

Direct System.

Residences, schoolrooms, offices, etc.....	60 to 80 cubic feet

Assembly halls, stores, factories, etc..... 75 to 100 cubic feet
Churches, large auditoriums, etc.....150 to 200 cubic feet

For rooms exposed to the outside air, such as front rooms, 1 to 70 is about the proper ratio; for inner rooms, 1 to 100 is sufficient. For large stores with buildings on each side, one square foot of radiating surface suffices for 100 cubic feet under the most severe conditions.

Dimensions of Steam Pipes.—For direct heating, 1 square inch of pipe area is allowed for every 100 square feet of radiating surface if the pipe is of medium length. For longer pipes the diameter of the main leading from the boiler to the various rooms can be calculated by this formula:

$$d = \frac{\sqrt{R + r}}{10}$$

d = diameter of main in inches.
R = square feet of radiating surface in the rooms.
r = square feet of surface in main pipe.

The surface r of the main can, of course, only be calculated after the diameter d is known. Since the surface of the main is very small compared to that of the radiators, its omission will not make any practical difference. Omitting this surface from the calculation, the formula reads:

$$d = \frac{\sqrt{R}}{10}$$

Example.—What must be the diameter of the main necessary for 2,500 square feet of radiating surface?

Solution.—$d = \frac{\sqrt{2,500}}{10} = \frac{50}{10} = 5.$

The diameter of the main must be 5 inches.

Pitch of Steam Pipes.—An important point to be observed in a direct heating system is that all pipes leading to and from the radiators have the necessary pitch, so that an effective drainage is maintained. If there is any knocking or pounding noticeable in the system it is due to improper drainage.

Air Relief Valve on Radiator.—It is customary to provide each radiator with an air relief valve, the best devices being automatic. In these valves, a small piece of carbon gradually

closes the opening of the valve while expanding under the influence of the heat of the steam, so that practiccally all the air is expelled by the steam filling the radiator. When the steam is turned off and the piece of carbon gets cold it contracts, and, opening the valve, it admits air into the system. The formation of a vacuum is thus prevented and proper drainage made possible.

In large systems, all the radiators are connected by small air pipes into one main pipe, from which the air is released by means of a vacuum pump. This arrangement avoids any possible disturbance caused by individual air valves.

Dimensions of Boiler.—The proper dimensions of the boiler necessary for a direct heating system have already been given under "Indirect Heating."

One-Pipe System.—The one-pipe system is now generally in use, the steam flowing in one direction and the condensed water in the other in the same pipe. If the boiler is used for heating only and stands lower than the heater, no mechanical appliance is necessary for bringing the condensed water back to the boiler, since the water in this case flows into a stand or riser pipe, which is extended at the bottom and delivers the water into the boiler below the water line.

If higher pressure is carried on the boiler than is used for heating, it is reduced by means of a reducing valve, and the stand-pipe is connected to a receiver, from where the boiler feed pump delivers the condensed water directly to the boiler. A steam trap may also be used for relieving the pipes of the condensed water, which is thus discharged into a receiver. Sometimes the receiver which receives the condensed water from the stand-pipe is provided with a steam trap. In that case a second tank is necessary for receiving the condensed water from the trap.

Hot Water Heating.

Hot water heating is based on the circulation of hot water, usually of a temperature of about 160° F., through a system of pipes and radiators in every respect similar to those employed in direct steam heating. No mechanical appliances are necessary to bring about circulation; the hot water coming from the boiler is lighter than the water which has already given off its

heat in the system and therefore it rises to the radiators, while the cold and consequently heavier water flows to the boiler.

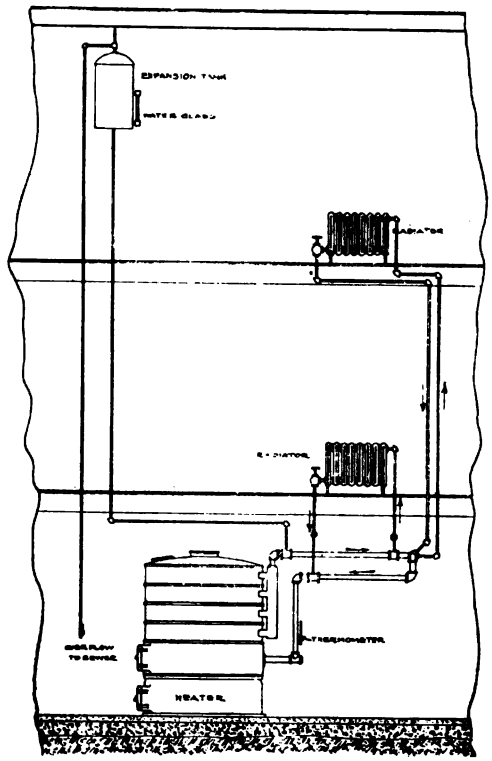


Figure 243. Hot Water Heating System.

The velocity of circulation depends mainly on the difference in temperature between the water leaving the boiler and that returning to it; it is furthermore influenced by the vertical distance of the radiators from the boiler, by the amount of friction in the system, change of direction of current, etc.

Dimensions and Arrangement of Pipes.—The diameter of the pipe leading to the radiators is governed by the amount of radiating surface, the height which the radiator stands above the boiler, and the number of changes in the direction of the current. Usually one square inch of cross-section of the supply pipe is allowed for every 50 to 100 square feet of radiating sur-

face. If the pipes are short, straight and more or less vertical, the proper ratio is about 1 to 100, if they are long or crooked and the system is low, then the ratio may be reduced to suit prevailing conditions, if necessary to 1 to 50.

All pipes rising from the boiler to the radiators should have a pitch of at least one inch in ten feet.

Expansion Tanks.—Every hot water heating system should be provided with an open expansion tank of at least one-twentieth the capacity of the entire system. Closed expansion tanks should be avoided.

Amount of Radiating Surface.—About 40% more radiating surface is required for hot water heating than where steam is employed. Otherwise the calculations are the same, and the formulas given in connection with steam heating may also be employed for hot water heating.

Heat Transmission Through Plates and Pipes.

The transmission of heat through iron or copper plates is the same for each degree difference in temperature, whether this difference is great or small, i. e. the transmission of heat is just as good with 1° F. difference as with 200° F. difference.

The transmission of heat from steam to water, through iron pipes, is about 120 B.T.U. per hour per square foot, for each degree difference of temperature, and for copper coils about 200 B.T.U. These are practical figures and should be used where ordinary and not ideal conditions exist.

Calculations along this line have very seldom been made by engineers, and the pipe surface necessary for heating water or other substances by steam has been taken more or less by guesswork, or figures were used which would apply only under ideal conditions, for instance, where absolutely clean and polished surfaces were employed and where there was perfect circulation of the liquids. In practice, however, such ideal conditions are not to be found, and figures based on actual practice such as the ones given above, should, therefore, be taken.

Coefficient of Heat Transmission.—By coefficient of heat transmission is meant the number of B.T.U. transmitted through one square foot of metal in one hour for each degree F. dif-

ference in temperature. The coefficient of transmission for steam to water through iron, for instance, is 120 B.T.U.

Coefficients of Heat Transmission Through Iron of Moderate Thickness.

Substances.	*Coefficient in B T.U.*
From combustible gases to water (steam boiler)	17.
From steam and hot water radiators to air for heating rooms	2.5
From steam to water at rest	120
From steam to boiling water	1,100
From water to water, flowing in opposite direction (counter current)	80.
From water to water, both being at rest (transmission varies from 10 to 80, according to circulation)	10.
From air to air	1.5

For copper surfaces which transmit heat more readily, 66% should be added to the above figures. For instance, if an iron surface transmits 2,000 B.T.U. in a certain time, a copper surface of the same dimensions will transmit $2{,}000 + 2{,}000 \times 0.66 = 3{,}320$ B. T. U. in the same length of time.

On the other hand, when the transmission through a copper surface is known, that through the same iron surface can be found by subtracting 40%. A copper surface for instance, transmits 5,000 B.T.U. in a given time. An iron surface of the same size will then transmit $5{,}000 - 5{,}000 \times 0.40$ or 3,000 B.T.U.

Example.—How many square feet of surface of iron pipe are necessary to heat the water in a cylindrical tank having a diameter of 12 feet and a height of 14 feet, from 50° F. to 190° F. in 6 hours, with exhaust steam at 2 pounds gauge pressure?

Solution.—The total weight of water to be heated is $12 \times 12 \times 0.7854 \times 14 \times 62.41 = 98{,}818$ pounds. The range of temperature through which the water has to be heated is $190 - 50 = 140°$ F., so that each pound of water must absorb 140 B.T.U. The total heat which must be furnished to the water by the exhaust steam, is, therefore, $98{,}818 \times 140 = 13{,}834{,}520$ B.T.U.

The difference in temperature between the exhaust steam inside of the coil and the water surrounding it, is in the be-

ginning 218° — 50° or 168° F., and after the water in the tank has been heated to 190° F. as required, the difference is 218 — 190 or 28° F. The average difference is, consequently, $\frac{168 + 28}{2}$ = 98° F.

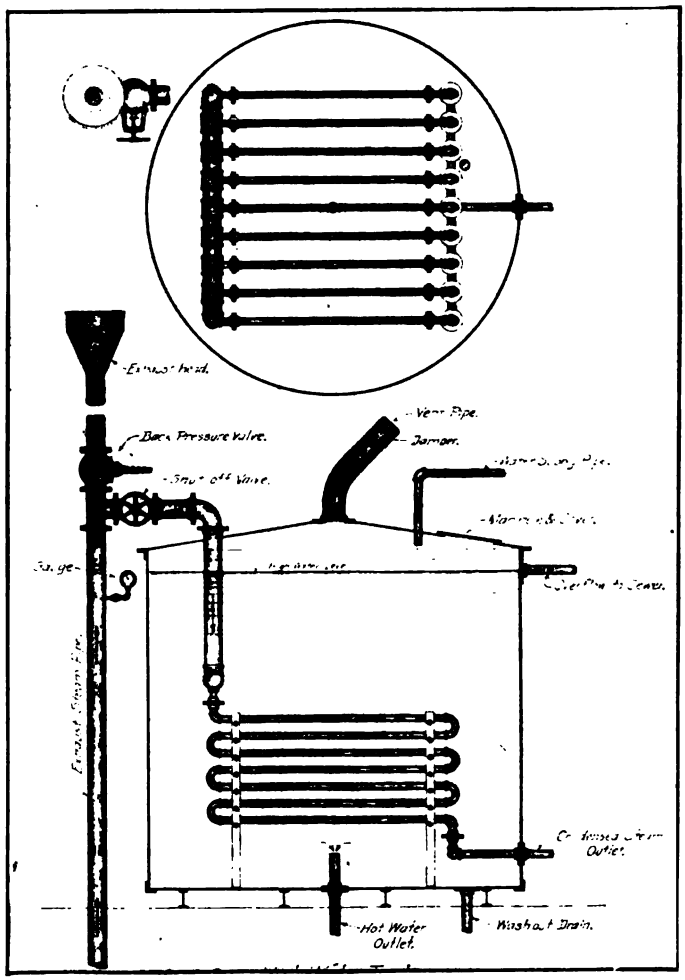


Figure 244. Hot Water Tank.

As one square foot of iron pipe surface will transmit 120 B.T.U. per hour per degree difference in temperature, a square foot can transmit for 98° F. difference and in 6 hours, $120 \times 98 \times 6 = 70{,}560$ B.T.U. It follows that the number of square feet necessary to transmit the total heat in the above example is:

$$\frac{13{,}834{,}520}{70{,}560} = 196 \text{ square feet.}$$

The above calculation can be expressed by the following formula:

$$S = \frac{Q \times W \times (t_1 - t)}{k \times H \times \left(t_2 - \frac{t_1 + t}{2}\right)}$$

S = square feet of pipe surface necessary.
Q = cubic feet of water to be heated.
W = weight in pounds of one cubic foot of water at temperature t.
t = initial temperature of water in degrees F.
t_1 = final temperature of water in degrees F.
t_2 = temperature of steam in degrees F.
H = number of hours in which water should be heated.
k = coefficient of heat transmission.

By substituting the figures of the above example for the letters in the formula, we obtain:

$$S = \frac{14 \times 12 \times 12 \times 0.7854 \times 62.41 \times (190 - 50)}{120 \times 6 \times \left(218 - \frac{190 + 50}{2}\right)} = 196$$

which means that 196 square feet of pipe surface are necessary.

Arrangement of Heating Coils.—Coils for the above purpose are generally made from 2-inch pipes with return bends. All the coils combined must have an area equal to that of the exhaust pipe entering the tank. The construction of such a tank is clearly illustrated in Figure 244.

Pipes and Pipe Fitting

Kinds of Pipe.

Materials.—The most common material at present used for water, gas and steam pipe is *mild steel,* this being cheaper than any other suitable material.

Wrought-iron pipe has certain advantages over mild steel pipe, such as that it is more easily bent, threads are more readily cut, and it has longer life under adverse conditions. However, wrought iron is not as strong as steel, and costs more; and hence, is used considerably less than steel.

Copper and *brass* are used occasionally, but, on account of their high cost, cannot compete with steel or wrought iron. Certain *alloys* have also been used for pipes, especially where superheated steam is used. The cost, however, makes their use limited.

Butt- and Lap-welded Pipe.—*Butt-welded* standard pipe is made in sizes up to and including 1¼ inch diameter. It is simply a flat strip of iron and the edges butted and welded together.

Lap-welded pipe is made from a strip, the edges of which are tapered so that when bent into a circle, these tapered edges overlap each other and give a greater contact surface than if butt-welded. For this reason lap-welded pipe is stronger than butt-welded, and can furthermore be more readily bent into coils without the seam or welded joint coming apart. Standard pipe of 1½ inch and over is all made lap-welded. Smaller lap-welded pipe is made to order.

Strength and Size of Pipe.—In regard to thickness of metal wall or strength, there are three classes of pipe, viz., *standard, extra strong* and *double extra strong.* Tables of sizes of these pipes are given on pages 747 and 748. It may be seen from the tables that all three classes have the same outside diameter for a given size of pipe; the inside diameter, of course, being smaller as the thicknsss of the metal increases.

The various sizes of iron or steel pipe are measured by the inside or *internal* diameter. The internal diameter, however, by which the pipe is designated, is not the *actual* internal diameter but the *nominal* diameter, the actual being always somewhat larger, as is shown in the pipe table.

Tubes are measured by the outside or external diameter.

The average factor of safety for steel or iron pipes is about 10. The pipes are subjected to hydrostatic tests at the mills as follows, the pressures being in pounds per square inch:

Standard butt-welded ⅛—3″.....600—1,000 lbs.
Standard lap-welded 3—12″.....500—1,000 lbs.
Extra heavy, butt-welded ⅛—3″.....600—1,500 lbs.
Extra heavy, lap-welded 1½—12″.....600—1,500 lbs.
Double extra heavy, butt-welded ⅛—2½″.....600—1,500 lbs.
Double extra heavy, lap-welded 1½—8″.....1,200—1,500 lbs.

Pipe Threads.

The ends of pipes used for screw connections are threaded according to Briggs or U. S. Standard system, the end of the pipe being tapered 1 to 32, so as to enable better connections. The table on page 747 gives the number of threads per inch for various sizes of pipe.

Cutting and Threading Pipe.—The ordinary tools used for cutting and threading pipes are too well known to require description. In cutting smaller pipe, the precaution should be observed that enough oil is used and that the cutter wheel is not tightened too quickly, as this would pinch off the pipe instead of cutting it.

Figure 245 shows a pipe properly cut, giving a full opening equal to the internal diameter. The Figure also shows the evil resulting from too rapid cutting or pinching off. The illustration shows an internal burr or ridge caused either by too much pressure on the cutter wheel, or by the use of a dull wheel.

Here it may be seen that the effective internal diameter is much less than the diameter of the pipe.

It may happen that the cutter wheel is dull, and that the formation of this burr is unavoidable, in which case the burr should be reamed out with a reamer specially made for the purpose, or with the pointed handle end of a large file.

In threading a piece of pipe, care should be taken that the chips do not clog the die, as it is this clogging which causes the thread to be "stripped" partly, or totally torn off. These chips can be thrown out by reversing the die one or two threads and giving the pipe a sudden, sharp rap or blow.

The thread or end of pipe should be well oiled by squirting oil into the die, as a dry die will wear quickly and cause the stripping of thread much more readily than one well supplied with oil. Soap suds, or a solution of soap in water, is cheaper, and is at the same time a good lubricant.

Figure 245. Correct and Incorrect Method of Cutting and Threading Pipe.

Cements for Thread Joints.—The usual cement used by gas fitters is white lead ground in oil to the consistency of a paste. This is for rough work, especially large pipe work. For smaller pipe, gas fitters use a regular gas fitters' cement, consisting of resin, shellac, etc., in fact, a cheap grade of sealing wax. In order to apply this, the fitting and pipe threads must first be heated so as to melt the cement, and the joint must be made while the fitting is hot enough to keep the cement in a melted state.

Steam fitters use mostly red lead in oil, but where an extra tight joint is wanted they often use litharge cement. This cement is made by rubbing litharge, or yellow oxide of lead, to a thick paste with concentrated glycerin, which produces, when it "sets" or dries (6 to 12 hours), a cement of almost metallic hardness.

In water pipe fitting the lead cements can be used, but if the threads are sound some fitters do not use any cement, figuring that when the water is turned on, any small leaks will "rust shut."

Kinds of Fittings.

The most common of the regular fittings used are illustrated by Figure 246. These fittings are made in three kinds, cast iron, malleable iron and galvanized iron.

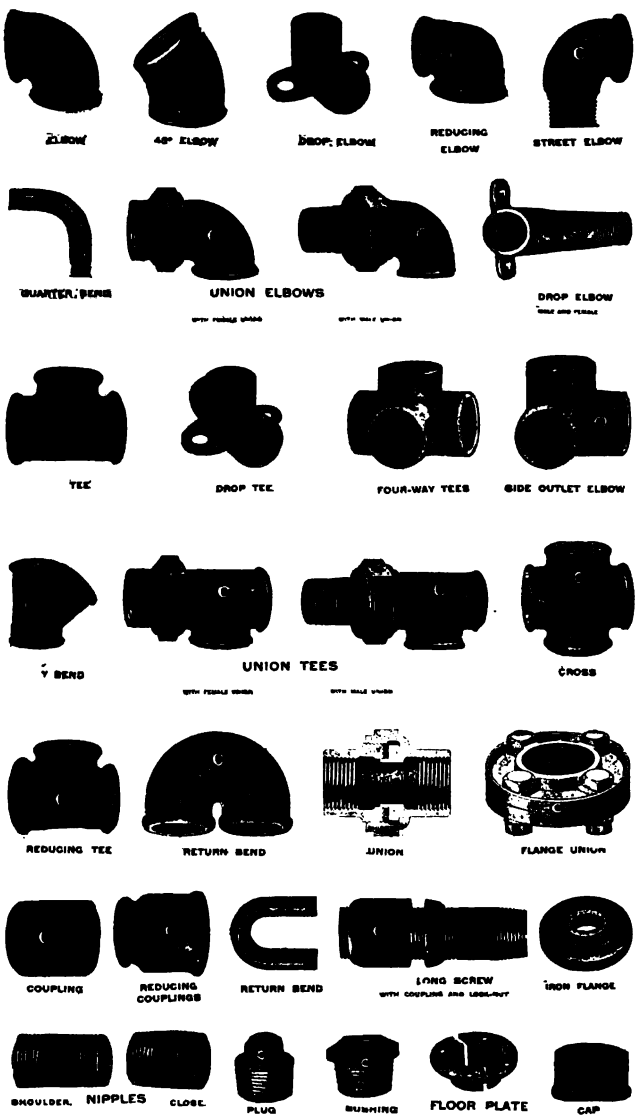


Figure 246. Common Fittings

Cast iron fittings are brittle, and are mostly used on heavy work. They are the cheapest of the three kinds.

Malleable iron fittings are a special kind of cast iron fittings which have undergone an annealing process by having been heated to red heat, and cooled gradually in 5 to 7 days. Malleable iron is tough, not brittle like cast iron, and malleable fittings can be struck with a hammer without breaking; furthermore, they are usually made thinner, consequently lighter, than cast fittings.

Galvanized fittings are usually malleable fittings coated with a covering of zinc to prevent rusting in damp places.

Most of the fittings shown in the illustrations require no description, those having special features being the following:

Drop fittings are such as have two "wings" or lugs with screw holes for fastening them to a ceiling or wall.

Street elbows are regular elbows having one of the ends with a male thread.

A male thread is one which is on the *outside*, as on a piece of pipe; a **female thread** is one on the *inside* of a pipe or fitting.

Unions are flange couplings in which the clamping together of the flanges is effected by a screw collar instead of by bolts. Unions are used mostly on smaller pipe, while flange joints are used on larger sizes.

Tees.—In designating the sizes of the openings in tees, the *run* is first mentioned and then the *outlet*. Thus in Figure 247 the run is reduced from ½-inch to ⅜-inch while the outlet is ½-inch. Such tees are called *reducing tees*, and the dimensions should be given when ordering, as a ½ × ⅜ × ½ reducing tee. If this is not correctly given, say ½ × ½ × ⅜, it would mean a tee with ½-inch run and ⅜-inch outlet.

Crosses (Figure 247) are designated like tees, the run may differ but the two outlet sizes are always the same.

Long screws or **running thread couplings** are used to connect the ends of two pipes already in place. A long screw is shown in the illustration of fittings, Figure 246 (second last row), but its operation is shown in Figure 248. Here it is desired to connect the pipes A and B with the long screw C. The first operation is to connect the long screw C with B by means of

coupling F, which can be done by bending either A or B a trifle to clear the other end of C. The coupling D is next turned to the left until it is tight upon pipe A, as shown at H in the second or center illustration. This will generally allow a leak at P, as the coupling fits somewhat loosely. This leak can then be stopped by winding some packing, like candle wicking soaked in tallow, white lead or other cement around the thread at P and then screwing lock-nut E to the left, to tighten this packing, as shown by K in the third or bottom part of Figure 248, the compressed packing being shown by P^1.

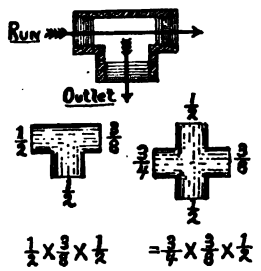


Figure 247. Tees and Crosses.

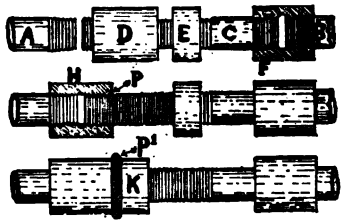


Figure 248. Long Screw.

Right and left nipples and couplings are, like long screws and unions, also used to connect the ends of two pipes already installed. In connecting with such a nipple or coupling it should be so done that both threads bind at the same time. As threads differ, it usually happens that one end of the nipple or coupling binds or "shoulders" before the other, leaving the latter loose and leaky.

This can be prevented by first screwing the nipple or coupling tight on one thread only, and noting the number of turns necessary to do this; then unscrewing this end and repeating the operation with the other thread. If it was noted, for instance, that one took 4 turns and the other 5½, all that is then necessary is to give the 5½-turn thread a 1½ turn before starting the second. An additional 4 turns will then bind or tighten both threads at the same time.

Expansion Joints.

When larger pipes, like steam mains between rigid walls, etc., are to be run, provision should be made for expansion. This

can be done by having several elbows so that the short pipes at an angle will bend or turn slightly to allow for this expansion, as in the two illustrations to the left of Figure 249 (the two views representing a quarter turn). Here the main pipe A in expanding will turn pipe B on C as a pivot.

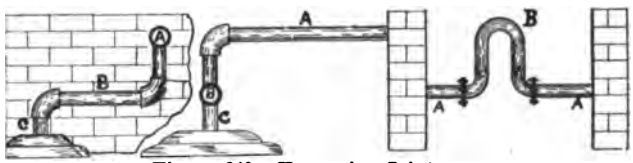


Figure 249. Expansion Joints.

On the right of Figure 249 is shown another expansion joint. Here the expansion of main pipe A is taken up by the inverted U-shaped tube B, which, being made of copper, will readily bend.

Flanged Fittings.

In larger pipe fitting, or where great strain is to be withstood, flange joints are used instead of threaded joints. Ammonia joints are almost exclusively flanged.

Figure 250. Flanged Coupling.

Figure 251. Flanged Coupling.

There are a number of different kinds of flanges in use. The commonest type, called flat-faced flange, is shown in Figure 250. Here the pipes are screwed into the flange, a gasket placed between and the two ends held together by bolts. This type under high pressure is apt to leak.

Instead of flat-faced flanges, so called *male and female* may be used. Figure 251 shows a standard make of this type, in which the gasket of lead or rubber is clamped into the confined space formed by the groove, in which it is compressed to form a joint. Male and female flanges are commonly employed in

ammonia fitting. A modification of this type is the tongued and grooved, in which a very narrow gasket is employed.

One of the best joints is the so-called lap-joint. In this type, the flanges are loose, which allows much flexibility.

Pipes may also have a *welded flange,* and though expensive this joint is the most reliable, especially where high pressures or high temperatures are involved. The faces on this flange are generally raised a slight amount and ground so that a thick gasket is not necessary.

Figure 252.
Globe Valve.

Figure 253.
Check Valve.

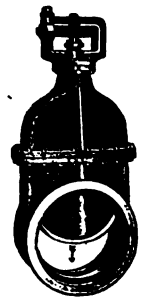


Figure 254.
Gate Valve.

Valves.

The valves in general use can be divided into three kinds: globe, check and gate or straight-way valves.

Globe valves (Figure 252) are of two kinds: straight and angle valves.

Check valves (Figure 253) allow the passage of water, steam, etc., *one way only* while the passage through the other or reverse way is checked by an automatic valve. In Figure 253, V is the check, which rests upon its seat by gravity, as this seat is slightly on the slant. When water, etc., enters in the direction of the arrow B, the valve V lifts and opens, while if the water enters in the direction of arrow A it presses upon the valve V and keeps it closed.

Gate or straight-way valves (Figure 254) open and shut by means of a disc or "gate." Very large valves are usually of this kind. Gate valves have the advantage that they give a straight full opening.

Installation of Pipe Systems.

In installing steam and water systems, the pipes should always be laid with a *pitch* or *slant,* so that they can be drained. A *vent* cock should be placed at the highest, and a *drip* cock at the lowest point of the system, in order to allow the water of condensation, etc., to flow out. In the laying of pipes, if the formation of a "pocket" or low point is unavoidable, this low point, or pocket, should also be supplied with a drip cock.

In gas piping, this is unnecessary, except where the gas pipe is *exposed to the cold.* At this point, drips should also be inserted because it will be found that in cold weather a liquid deposits which will cause the lights to "go out" unless this liquid is removed from time to time. This liquid does not, as is generally supposed, consist entirely of water, but of the heavier and less volatile hydrocarbons contained in the gas, which condense or liquefy at lower temperatures.

Air Line Piping.

The placing of drip cocks is also necessary in air lines, because here considerable water is deposited. This is due to the fact that when air is compressed it becomes warm or hot, and can hold more moisture than when it is cold. As this hot compressed and moisture-laden air is forced into the air line, it cools and deposits much of the moisture as water.

In air lines, it is not only necessary to have drips but also traps or reservoirs at low points, so that these drips need only be opened from time to time.

Methods of Testing for Leaks.

The usual method of testing a pipe system for leaks is by means of air or water pressure, principally the former. Before the "proving" or testing pump is attached, it should be ascertained that all openings in the piping are closed or "capped up."

The pump is then attached and air pumped into the system until the mercury column shows the desired height in the gauge glass, in case a gas fitters' proving pump is used, or until the pressure gauge shows the desired amount of pressure. The valve or cock between the pump and mercury column or gauge is then

closed and the system allowed to remain under this pressure for some time.

Should the system be tight, the mercury column or gauge will remain at the same height or pressure, but if there is a leak, the mercury or gauge pressure will fall with a rapidity corresponding to the magnitude of the leak.

Should a leak be indicated, there are two common methods of finding it. Where the pipe is all exposed, and where the leak is a large one (indicated by the mercury or pressure falling almost immediately), the air should be again pumped in, after which, by running a lighted candle along the joints, the leak can usually be discovered by the flickering of the light when opposite the leak. If the leak is small, the usual method is to swab the pipe or joints with thick soap suds, after which any leaks will show by the formation of a globular soap bubble at the leak. For this purpose, instead of soap suds, brewers' finings are an excellent substance.

The second method for discovering leaks is to let out all the pressure and inject volatile strong smelling liquids, like sulphuric ether, into the pipe and again pump air into the system. This pumping forces the ether vapors to all parts of the system and causes it to escape through the leaks to the outside, where these leaks can then be located by the strong odor of the escaping ether vapors.

Caution.—In using ether, care should be taken to keep any flame away from it, because ether is much more inflammable than gasoline and benzine.

Oil of peppermint mixed with hot water is sometimes used for large upright pipes, like vent pipes to the roof. By pouring this mixture into the pipes any leak can be located by the strong odor of peppermint. Here two men are required, because the man who pours in the mixture should stay outside, inasmuch as one drop on his hands or clothes, if he comes inside, will give a wrong scent.

When water pressure is used, all that is necessary is to follow up the system and observe where the water either squirts or oozes out.

Wrought Iron and Steel Steam, Gas and Water Pipe.—Table of Standard Dimensions.*

Diameter.			Thickness.	Circumference.		Transverse Areas.			Length of Pipe per Square Foot of		Length of Pipe containing One Cubic Foot.	Weight per Foot of Length.	No of Threads per Inch of Screw.	Contents in Gals. per Foot of Length	Weight of Water per Foot of Length
Nominal Internal.	Actual External.	Actual Internal.		External.	Internal.	External.	Internal.	Metal.	External Surface.	Internal Surface.					
Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Sq. Inch.	Sq. Inch.	Sq. Inch.	Feet.	Feet.	Feet.	Lbs.		Gallons.	Lbs.
⅛	.405	.27	.068	1.272	.848	.129	.0573	.0717	9.432	14.15	2513.	.241	27	.0030	.0247
¼	.54	.364	.088	1.696	1.144	.229	.1041	.1249	7.074	10.49	1383.3	.42	18	.0054	.0451
⅜	.675	.494	.091	2.121	1.552	.358	.1917	.1663	5.659	7.73	751.2	.559	18	.0100	.0831
½	.84	.623	.109	2.639	1.957	.554	.3048	.2492	4.547	6.13	472.4	.837	14	.0158	.1321
¾	1.05	.824	.113	3.299	2.589	.866	.5333	.3327	3.638	4.636	270.	1.115	14	.0277	.2311
1	1.315	1.048	.134	4.131	3.292	1.358	.8626	.4954	2.905	3.645	166.9	1.668	11½	.0449	.3738
1¼	1.66	1.38	.14	5.215	4.335	2.164	1.496	.668	2.301	2.768	96.25	2.244	11½	.0778	.6483
1½	1.9	1.611	.145	5.969	5.061	2.835	2.038	.797	2.010	2.371	70.66	2.678	11½	.1060	.8831
2	2.375	2.067	.154	7.461	6.494	4.43	3.356	1.074	1.608	1.848	42.91	3.609	11½	.1746	1.4543
2½	2.875	2.468	.204	9.032	7.753	6.492	4.784	1.708	1.329	1.548	30.1	5.739	8	.2489	2.0731
3	3.5	3.067	.217	10.996	9.636	9.621	7.388	2.243	1.091	1.245	19.5	7.536	8	.3867	3.2015
3½	4.	3.548	.226	12.566	11.146	12.566	9.887	2.679	.955	1.077	14.57	9.001	8	.5144	4.2847
4	4.5	4.026	.237	14.137	12.648	15.904	12.73	3.174	.849	.949	11.31	10.665	8	.6622	5.5163
4½	5.	4.506	.246	15.708	14.162	19.635	15.961	3.674	.764	.848	9.02	12.34	8	.8303	6.9164
5	5.563	5.047	.259	17.477	15.849	24.306	19.99	4.316	.687	.757	7.2	14.502	8	1.0399	8.6623
6	6.625	6.065	.28	20.813	19.054	34.472	28.888	5.584	.577	.63	4.98	18.762	8	1.5027	12.5181
7	7.625	7.023	.301	23.955	22.063	45.664	38.738	6.926	.501	.544	3.72	23.271	8	2.0152	16.7865
8	8.625	7.982	.322	27.096	25.076	58.426	50.04	8.386	.443	.479	2.88	28.177	8	2.6031	21.6840
9	9.625	8.937	.344	30.238	28.076	72.76	62.73	10.03	.397	.427	2.29	33.701	8	3.2633	27.1830
10	10.75	10.019	.366	33.772	31.477	90.763	78.839	11.924	.355	.381	1.82	40.065	8	4.1013	34.1636
11	12.	11.25	.375	37.699	35.343	113.098	99.402	13.696	.318	.339	1.456	45.95	8	.....	.....
12	12.75	12.	.375	40.055	37.7	127.677	113.098	14.579	.300	.318	1.27	48.985	8	.....	.....
13	14.	13.25	.375	43.982	41.626	153.938	137.88	16.051	.273	.288	1.04	53.922	8	.....	.....
14	15.	14.25	.375	47.124	44.768	176.715	159.485	17.23	.255	.268	.903	57.893	8	.....	.....
15	16.	15.25	.375	50.265	47.909	201.062	182.655	18.407	.239	.250	.788	61.77	8	.....	.....
....	18.	17.25	.375	56.549	54.192	254.47	233.706	20.764	.212	.221	.616	69.66	....	.....	.....
....	20.	19.25	.375	62.832	60.476	314.16	291.04	23.12	.191	.198	.495	77.57	....	.....	.....
....	22.	21.25	.375	69.115	66.759	380.134	354.657	25.477	.174	.179	.406	85.47	....	.....	.....
....	24.	23.25	.375	75.398	73.042	452.39	424.558	27.832	.159	.164	.339	93.37	....	.....	.....

* The figures for amount of water in pounds and gallons are based on a weight of 62.4 pounds per cubic foot and 8.33 pounds per gallon.

Wrought Iron Welded Extra Strong Pipe. —Table of Standard Dimensions.

Diameter.			Thickness.	Nearest Wire Gauge.	Circumference.		Transverse Areas.			Length of Pipe per Square Foot of		Nominal Weight per Foot.
Nominal Internal.	Actual External.	Actual Internal.			External.	Internal.	External.	Internal.	Metal.	External Surface.	Internal Surface.	
Inches.	Inches.	Inches.	Inches.	No.	Inches.	Inches.	Sq. Inches.	Sq. Inches.	Sq. Inches.	Feet.	Feet.	Pounds.
¼	.405	.205	.1	12½	1.272	.644	.129	.033	.096	9.433	18.633	.29
⅜	.54	.294	.123	11	1.696	.924	.229	.068	.161	7.075	12.980	.54
½	.675	.421	.127	10½	2.121	1.323	.358	.139	.219	5.657	9.07	.74
¾	.84	.542	.140	9	2.639	1 703	.554	.231	.323	4.547	7.046	1.09
1	1.05	.736	.157	8¼	3.299	2.312	.866	.452	.414	3.637	5.109	1.39
1	1.315	.951	.182	7	4.131	2.988	1.358	.71	.648	2.904	4.016	2 17
1¼	1.66	1.272	.194	6¼	5.215	3 996	2.164	1.271	.893	2.301	3 003	3.
1½	1.9	1.494	.203	6	5.969	4.694	2.835	1.753	1.082	2.01	2.550	3.63
2	2.375	1 933	.221	5	7.461	6 073	4.43	2.985	1.495	1.608	1.975	5.02
2½	2.875	2.315	.28	2	9.032	7.373	6.492	4 209	2.283	1.328	1.649	7.67
3	3.5	3.892	.304	1	10 996	9.085	9.621	6.569	3.052	1.091	1.328	10.25
3½	4.	3 358	.321	0	12.566	10.549	12.566	8.856	3.71	.955	1.137	12.47
4	4.5	3 818	.341	0	14.137	11.995	15.904	11.449	4.455	.849	1.	14.97
5	5.563	4.813	.375	00	17.477	15.120	24.306	18.193	6.12	.687	.793	20.54
6	6.625	5.75	.437	000	20 813	18.064	34.472	25.967	8.505	.577	.664	28 58

Double Extra Strong Pipe.

¾	.84	.344	.298	1	2.639	.766	.554	.047	.507	4.547	15.667	1.7
¼	1.05	.422	.314	1	3.299	1.320	.866	.139	.727	3 637	9.049	2 44
1	1.315	.587	.364	00	4.131	1.844	1.358	.271	1.087	2.904	6.508	3.65
1¼	1.66	.885	.388	00	5.215	2.78	2.164	.615	1.549	2.304	4.317	5 2
1½	1.9	1.088	.406	000	5.969	3.418	2.835	.93	1.905	2.01	3.511	6.4
2	2 375	1 491	.442	0000	7.461	4.684	4.43	1.744	2.686	1.608	2.561	9.02
2½	2.875	1.755	.560	⅛—	9.032	5.513	6.492	2.419	4.073	1.328	2.176	13.68
3	3.5	2.284	.608	¼—	10.996	7.175	9.621	4.097	5.524	1.091	1.672	18.56
3½	4.	2.716	.642	⅜—	12.566	8.533	12.566	5.794	6.772	.955	1.406	22.75
4	4 5	3.136	.682	½—	14.137	9.852	15 904	7.724	8.18	.849	1.217	27.48
5	5.563	4.063	.75	⅝	17.477	13.764	24.306	12.965	11.34	.687	.940	38 12
6	6.625	4.875	.875	¾	20 813	15.815	34.472	18 666	15.806	.577	.784	53.11

Iron Pipe Size Brass Tubes.

Made to correspond with Iron Tubes, and to fit Iron Tube Fittings.

List of Sizes, Lengths, Etc.

Outside Diameter.	Same as Iron Size.	Weight Per Foot.	
		Brass.	Copper.
13-32 inch.	1-8 inch.	.30 pounds.	.31 pounds.
9-16 "	1-4 "	.43 "	.45 "
11-16 "	3-8 "	.58 "	.61 "
13-16 "	1-2 "	.80 "	.84 "
1 1-16 "	3-4 "	1.17 "	1.23 "
1 5-16 "	1 "	1.67 "	1.75 "
1 5-8 "	1 1-4 "	2.42 "	2.54 "
1 7-8 "	1 1-2 "	2.92 "	3.07 "
2 3-8 "	2 "	4.17 "	4.38 "
2 7-8 "	2 1-2 "	5. "	5.25 "
3 1-2 "	3 "	8. "	8.40 "
4 "	3 1-2 "	10. "	10.50 "
4 1-2 "	4 "	12. "	12.00 "

Seamless Drawn Brass and Copper Tubes.

(Pounds per Lineal Foot)

The upper figures denote copper, the lower brass, tubes.

Thickness, Decimal of an Inch.	Fraction of an Inch (Closely).	Gauge Number.	Outside Diameters in Inches. Brown & Sharpe or American Gauge.									
			½	¾	1	1¼	1½	2	2½	3	4	5
0.081	5-64	12	0.41 0.39	0.65 0.62	0.90 0.86	1.14 1.09	1.39 1.32	1.88 1.79	2.37 2.26	2.86 2.72	3.84 3.66	4.82 4.59
0.064	1-16	14	0.34 0.32	0.54 0.51	0.69 0.65	0.88 0.83	1.06 1.00	1.43 1.34	1.80 1.69	2.28 2.17	3.06 2.91	3.83 3.65
0.051	.....	16	0.27 0.26	0.43 0.41	0.59 0.56	0.73 0.70	0.89 0.85	1.20 1.14	1.51 1.44	1.82 1.73	2.44 2.32	3.04 2.90
0.045	3-64	17	0.25 0.24	0.39 0.37	0.52 0.50	0.66 0.63	0.80 0.76	1.07 1.02	1.34 1.28	1.62 1.54	2.16 2.06	2.72 2.59
0.040	.....	18	0.22 0.21	0.35 0.33	0.47 0.45	0.59 0.56	0.71 0.68	0.96 0.91	1.20 1.14	1.45 1.38	1.93 1.84	2.43 2.31
0.036	.....	19	0.20 0.19	0.31 0.30	0.42 0.40	0.52 0.50	0.64 0.61	0.85 0.81	1.07 1.02	1.29 1.23	1.72 1.64	2.15 2.05
0.032	1-32	20	0.18 0.17	0.27 0.26	0.38 0.32	0.47 0.45	0.57 0.54	0.77 0.73	0.96 0.91	1.14 1.09	1.53 1.46	1.92 1.83
0.285	.....	21	0.16 0.155	0.25 0.24	0.34 0.32	0.42 0.40	0.50 0.48	0.68 0.65	0.85 0.81	1.02 0.97	1.36 1.30	....
0.053	.....	22	0.146 0.139	0.226 0.211	0.30 0.285	0.37 0.358	0.45 0.43	0.61 0.58	0.77 0.73	0.91 0.87	1.22 1.16	....

Friction and Lubricants

Friction has already been defined on page 24, and as stated there, two kinds of friction must be dealt with, namely, *sliding* and *rolling* friction. In that chapter, the term coefficient of friction is explained, and a table of friction coefficients for various bodies is given.

The losses of mechanical energy resulting from friction are considerable, and the question of reducing them to a minimum has been and is still being given much study and experimentation by both scientists and practical engineers. Friction is, on the other hand, used for transmitting or transferring power or motion from one moving body to another, for instance, in the numerous different styles of belt and rope drives, the friction clutches and the movement of trains, cars, automobiles, etc. This chapter, however, will deal only with the friction which causes loss in power.

In lubricating practice, there must be considered the friction of solids and the friction of liquids. When the surfaces of two solids move against each other, their motion is retarded by friction. The smoother these surfaces are, the less friction caused by the interstices or high points in the surfaces of the one body sinking into the depressions in the other, will take place. The energy utilized for overcoming friction is transformed into heat, and in the case of metal bearings the heat causes the bodies to expand, which usually brings the surfaces still more closely together, thereby producing greater friction and more heat. Such friction, even if very small at the beginning, rapidly increases, and soon can become so great that the loss of mechanical energy is enormous. If the bodies are of a different degree of hardness, the softer will suffer in wear the most on account of the friction.

The friction of liquids is small as compared to that of solids, which is due to the fact that they flow easily, and are retarded

only very slightly in their movement by the irregularities or projections on the surfaces.

Theory of Lubrication.

In order to prevent the surfaces of two solid bodies in motion from coming in direct contact with each other, a lubricant is introduced between them. The action of this lubricant is to reduce friction, which it accomplishes not alone by filling the interstices or irregular points of such surfaces, but also by keeping them apart. It also takes up some heat that may be generated, and carries it away from the wearing surfaces. Although the nature of the lubricant employed for preventing friction is of great importance, its application is equally vital, because even the best lubricant will be ineffective in its function, if not applied in such a way that it fills in between the moving surfaces at every place. Since the most commonly employed lubricants are oils, it is the general practice, wherever possible, to flood the wearing surfaces with oil, so as to be sure that all points where friction occurs are lubricated, or that there is a thin film of oil between them. Where the flooded system cannot be applied, a certain quantity of the lubricant is allowed to enter at regular and frequent intervals by means of a suitable mechanical device.

If the lubricant is properly applied, the possible friction between the moving surfaces of the solids will be almost completely done away with, and the only friction that will occur will be the friction from the lubricant itself.

As far as the properties of liquid lubricants are concerned, the main one is the *viscosity*, or the power of the liquid particles to adhere to each other. Water is a good example of a liquid having a low viscosity, because the particles have a tendency to separate readily; for instance, if water is allowed to fall on the floor in the form of a thin stream, the drops will splash in all directions. When this is done with an oil, it will not act in that way, but the particles will cling together, in other words they will show cohesion. A liquid possessing cohesion to a marked degree is also usually of good adhesion, namely, it will stick to surfaces of other substances with which it comes in contact.

Besides this property of viscosity, a good lubricant must not volatilize at too low temperatures and it must have no detri-

mental action upon the materials of which the moving parts are made.

Lubricants.

With a few exceptions all of the lubricants commonly employed are oils or greases. Animal and vegetable oils and fats were used years ago, but have been replaced almost entirely by petroleum or the so-called mineral oils. The principal *advantages of mineral over animal and vegetable oils* are the following:

1) Their cheapness.

2) Being hydrocarbons, they do not decompose or oxidize upon exposure to air, which would make them liable to become gummy, rancid or thick; and when properly refined they will have no corrosive action on metals.

3) By means of fractional distillation, they can be separated into a great many different grades of oil, which differ principally as to the viscosity and flash test.

4) Being non-oxidizable, there is no danger of spontaneous combustion.

The ideal oil under any circumstances is the one that possesses the greatest adhesion to surfaces and at the same time has the least cohesion among its own particles.

The conditions under which the wearing surfaces of machinery are operated differ greatly, and the properties of the lubricating oil employed must be such as will not alone meet the requirements as to adhesion and cohesion, but also will not exercise a detrimental effect upon the materials of which the wearing surfaces are constructed. Different types of oil are on the market, and these vary from the lightest spindle oil to the dense, heavy, sometimes semi-solid cylinder oils. The type of oil to be employed as a lubricant is determined by the *speed*, the *pressure*, the *temperature*, the *material*, and *condition* of the wearing surfaces.

Kinds of Lubricants.

The kinds of lubricants employed may be divided into three general groups:

a) Mineral oils.

b) Animal and vegetable oils and fats.

c) Solid lubricants.

Mineral Oils.

Crude petroleum, which is the source of the mineral lubricating oils, is a highly complex substance consisting of a large number of different hydrocarbons. In former years, some varieties of the crude oil were frequently used in the raw state for lubricating heavy bearings. Such oils were designated as natural oils, and were usually purified by sedimentation, straining, and then filtration through boneblack, in order to remove any tarry or other substances in suspension. The employment of natural oils, however, is seldom resorted to nowadays.

So-called *reduced oils,* which are obtained by distilling petroleum with steam while under pressure or under a vacuum until the lighter oils have been removed, are employed chiefly for lubricating heavy pressure and slow-speed bearings. The removal of the lighter hydrocarbons in a way concentrates the petroleum, whereby the viscosity of the reduced oil is greatly increased.

The removal of these lighter hydrocarbons is sometimes carried out in open shallow tanks, in which the petroleum is exposed to sunlight while floating upon water warmed by steam. Such oil is called *sunned oil.*

The lubricating as well as illuminating and power-producing oils, found on the market today, are obtained by a process of distillation, during which the petroleum is heated at gradually increased temperatures, and the resulting vapors distilled and kept separate from each other.

The various distillates are purified by treating with sulphuric acid in order to bleach them, and the acid is then removed by washing the oils in a caustic soda solution before redistilling them and obtaining the various types of commercial oils. Filtering through boneblack (animal charcoal) is also resorted to when oils of a pale color are desired.

When subjected to fractional distillation, the petroleum yields three different distillates:

1) The *naphtha distillate,* from which, when further fractionated, the following commercial hydrocarbons are obtained: rhigoline, petroleum ether, gasoline, naphtha and benzine.

2) The *burning oil distillate,* which is separated by further

distillation into different grades of kerosene or petroleum and lantern oils.

3) The *lubricating oil distillate* is obtained when the residium of the burning oil distillate is distilled by means of superheated steam and yields the so-called "neutral," "spindle," "loom," "engine" and "cylinder" oils.

After obtaining the lubricating oil distillate, the residue is further treated, and paraffin wax and vaseline extracted. The final residue is placed on the market as petroleum asphalt.

Animal and Vegetable Oils and Fats.

Oils and fats derived from animal or vegetable tissues are usually called *fixed oils*. Fats are solid at ordinary temperatures and oils are liquid. Fats, however, become oils when heated, and fixed oils become fats when cooled. They possess the characteristic property that they readily take up oxygen from the air. Mineral oils differ radically from fixed oils in this respect, and are not affected by the air. Some of the fixed oils, for instance, linseed oil, when oxidized, become a resin-like substance, and such are used chiefly in the manufacture of paints. They are known as *drying oils,* and are unfit for lubricating purposes.

This absorption of oxygen by fats and oils is accompanied by a rise in temperature, the more so, the larger the surface of the oil exposed to air. If rags or machinists' waste, saturated with fixed oils, especially where placed on a pile or enclosed in a box, are allowed to lie around, they may take up oxygen to such an extent as to ignite, namely undergo spontaneous combustion.

Another characteristic property of the fixed oils and fats is that they saponify when treated with an alkali, forming soaps. This saponification allows fixed oils to be distinguished from mineral oils, which are not acted upon by alkalies.

The fixed oils commonly employed for lubricating purposes, sometimes alone, but generally mixed with mineral oil, are the following:

Animal origin.	*Vegetable origin.*
Tallow oil, Neatsfoot oil,	Rape seed oil,
Lard oil, Whale (sperm) oil,	Castor oil,
Tallow, Lard.	Corn oil.

Blown or Thickened Oils.

Blown rape and cottonseed oils are sometimes mixed with mineral oils and used for lubricating purposes. The oil which is to be blown is placed in a jacketed kettle, heated to 170 to 230° F., and a current of air is passed through it. As a result the oil becomes thick and of very high viscosity.

Greases.

Greases have been termed plastic lubricants, and usually represent mineral oils that have been thickened with soda, lime or aluminium soap, or a solid lubricant, until a semi-solid, gelatinous mass has been obtained. They are generally used where heavy pressure, low-speed lubrication is desired, for instance, on heavy engine bearings, crank pins, pillow blocks, shaft bearings, on the curved rails of street railway tracks, in gear cases, for lubricating driving chains and for preventing friction and wear on wire cables and ropes. According to W. H. Davis, in his book on "Friction and Lubrication," greases have certain advantages in the way of cleanliness, ease of application, etc., yet on general mill and factory machinery the use of grease instead of oil will tend to increase the friction load.

Solid Lubricants.

The solid lubricants most commonly used are finely powdered talcum (soapstone) and graphite. Of these, graphite possesses the greatest lubricating properties and is almost exclusively used when applying a solid lubricant.

Among the different grades of lubricating graphite, the "Acheson Graphite" is worthy of special mention. This is an almost chemically pure, deflocculated graphite, which, when worked into the form of a paste with lubricating oil, is sold under the trade name "Oildag," and when mixed with grease as "Gredag." On account of the very finely divided condition of the graphite, it fills the interstices or rough points of wearing surfaces, and also coats them with a fine film, so that they become extremely smooth and polished, thereby reducing the friction to a minimum.

Physical and Chemical Properties of Lubricating Oils.

The suitability of a lubricating oil that is to be used for certain purposes depends to a great extent upon its properties, and

these can be determined by laboratory tests. The properties, both physical and chemical, generally considered in the valuation of lubricating oils in common use, are:

1) Specific gravity or Beaumé degrees.
2) Viscosity.
3) Flash test.
4) Fire test.
5) Cold test.
6) Admixture of animal or vegetable oils and fats.
7) Acidity or free acid.

Specific Gravity.—The specific gravity of an oil is determined either by weighing a measured quantity at a certain temperature or by means of a hydrometer, showing either the specific gravity directly or the degrees according to the Beaumé scale. For sufficiently fluid oils, the employment of a hydrometer is the simplest method. The oil to be tested must have the exact temperature required for the hydrometer, which usually is 60° F.

When determining the specific gravity of an oil gravimetrically, a 100 cubic centimeter flask, previously tested as to accuracy and weighed dry, is filled with oil having a temperature of 59° F. (15° C.), and the total weight determined. By subtracting the weight of the empty flask, the weight of 100 cubic centimeters of the oil is obtained. This figure divided by 100 gives the specific gravity. If the latter is known, the degrees Beaumé can be found by referring to the Beaumé table for liquids lighter than water.

Viscosity.—This very important property, which indicates the cohesive qualities of an oil, is determined by means of an instrument called a viscosimeter. There are numerous types of this instrument, among which are to be mentioned Traube's, Saybolt's, Tagliabue's, and Doolittle's torsion viscosimeters. The Tagliabue instrument is employed most commonly in commercial oil testing in this country. It consists of an oil chamber surrounded by a jacketed boiler, through which either water or steam may be passed. At the outlet of the oil chamber is a stop-cock to which a steel nipple is attached. The oil chamber is filled with about 70 cubic centimeters of the strained oil, and as soon as it has the desired temperature, namely, 70° F. for thin oils, and 212° F. for heavy oils, the stop cock is opened, and exactly 50 cubic centimeters of the oil allowed to run into a measuring flask.

The time in seconds is observed by means of a stop-watch, and this multiplied by two gives the viscosity.

Flash Test.—The flash test represents the temperature to which the oil must be heated, so that the vapors when mixed with air will momentarily ignite or flash when a flame is applied. It is usually determined by heating the oil, contained in a suitable metal cup, about 3° F. per minute until the flash is obtained. The temperature is then read on the thermometer (which registers up to 680° F.). The instrument commonly used is Tagliabue's open tester.

Fire Test.—When determining the flash test, the oil, upon further heating, will soon give off vapors to such an extent that if the flame is applied, they will catch fire and burn. The temperature at which this occurs is called the fire test, and it usually is 40 to 60° F. higher than the flash test.

Cold Test.—The temperature at which an oil congeals, or becomes so solid that it no longer flows, is termed the cold test or setting point. It is determined by placing a small bottle (about one inch in diameter) of the oil, in a freezing mixture of ice and salt. After the oil has solidified, the bottle is taken out and the oil allowed to warm slowly until it just about begins to flow. The temperature is then read on the thermometer inserted in the bottle.

Admixture of Animal or Vegetable Oils or Fats.—An accurate determination of an admixture of animal or vegetable oil or fat is more within the province of the chemist than the practical engineer. The method generally employed is as follows:

Twenty grams of the oil are weighed off in a dry Erlenmeyer flask, and 100 cubic centimeters of an alcoholic potassium hydrate solution added. The latter is made by dissolving 3 grams of potassium hydrate in 100 cubic centimeters of grain alcohol. The flask is then closed with a stopper, fitted with a condenser, placed over a small flame, and the mixture boiled for one hour. The contents of the flask are then transferred into a separating funnel and allowed to stand for one hour. Inasmuch as the mineral oil is heavier than the alcoholic solution it will settle to the bottom, and can then be run off. The alcoholic solution which remains is now mixed with about 25 cubic centimeters of distilled water, so as to dissolve the soap, and then dilute sulphuric acid (one

part acid to 10 parts of distilled water) added, until the mixture reacts distinctly acid when tested with blue litmus paper. It is then well shaken, and about 50 cubic centimeters of sulphuric ether added, which will dissolve the fatty acids liberated by the addition of the acid. The ether, being lighter specifically, will float on top, and the alcoholic water can then be drawn off. The ether is then run into a weighed Erlenmeyer flask or beaker, the separating funnel washed twice with more ether, and this also run into the flask, and after evaporating the ether on a water bath, the flask or beaker should be dried at 95° C. for one-half hour. The residue thus obtained represents the fatty acids of the fixed oil.

Acidity.—Free acids in a refined lubricating oil may be present as either free mineral acids, usually sulphuric acid, or free fatty acids, originating from animal or vegetable fats or oils used for compounding the mineral oil.

According to Archbutt & Deeley, the total acidity of an oil can be determined by dissolving 5 to 10 grams of the oil in a neutral mixture of ether (4 parts) and absolute alcohol (1 part) and titrating the solution with decinormal caustic soda solution, using phenolphthalein as an indicator.

The usual test for free mineral acids is to add about 10 cubic centimeters of the oil to 100 cubic centimeters of hot distilled water and thoroughly stir the mixture. Afer cooling, methyl-orange is added as an indicator, since it is not affected by fatty acids (of animal or vegetable origin). The presence of free mineral acids will cause the water to become *red*.

Lubrication of Machinery.

The selection of the lubricants best adapted for the numerous types of machinery operating under varying conditions is no easy matter. As a rule, the speed and pressure of the bearing surfaces are the main points considered. The temperature is also of great importance, especially in the selection of oils that are used at either very low or very high temperatures. Furthermore, if the proper results are to be obtained, the manner or means of application of the lubricant must be given due attention.

The speed and pressure of the bearing surfaces may be classified as follows:

1) Slow speed and heavy pressure.
2) Moderate speed and pressure.
3) High speed and light pressure.

For slow speed and heavy pressure, solid lubricants, greases or oils of heavy body, either pure mineral or compounded with tallow or lard, are used.

Where high speed and light pressure bearing surfaces are to be lubricated, a pure mineral oil of light body is to be preferred.

In ordinary engineering practice, the lubricating oils commonly employed are known as engine oils and cylinder oils. Numerous types of each kind are on the market, which differ as to their viscosity, cold, flash and fire tests, their suitability for certain machines depending on these properties.

Cylinder Oils.

Cylinder oils are oils of heavy body, and are nearly always *compounded,* that is, petroleum oils of high viscosity are mixed with 1 to 10 per cent and sometimes as much as 25 per cent of fixed oil or fat, usually an acidless tallow, tallow oil, lard or lard oil. Under average conditions, such a mixture is to be preferred for steam cylinders, because the fixed oil or fat readily mixes or emulsifies with the water always contained in varying amounts in the steam. The resulting emulsion readily adheres to the wet wearing surfaces, and in that way the mineral oil is spread so that it can exercise the proper lubricating action. A pure mineral oil does not adhere to wet surfaces, and such oils, providing the flash and fire tests are sufficiently high, are not used for steam cylinder lubrication, excepting when the steam is very dry. The more moisture present in the steam, the greater should be the addition of fixed oil or fat. The employment of too high a percentage of fixed oils or fats, however, is undesirable, especially if the fat is acid in reaction, since corrosion of the metal wearing surfaces occurs; also where the condensed water is returned to the boiler and where exhaust steam is used directly for heating the boiler feed-water, because the emulsified fixed oil or fat does not separate readily from the condensed water or exhaust steam, and the introduction into the steam boiler of such oils or fats is highly undesirable (see chapter "Boiler Water and its Treatment").

Analyses of about 100 samples of steam cylinder oils made in the laboratories of the Wahl-Henius Institute, Chicago, during the year 1912-1913, gave the following results:

	Average	Maximum	Minimum
Specific Gravity	0.905	0.934	0.886
Viscosity at 212° F. (Tagliabue Viscosimeter).	168°	240°	102°
Flash Test	520° F.	604° F.	401° F.
Fire Test	554° F.	662° F.	419° F.

With two exceptions, which were pure mineral oils, all of the oils were compounded with 1 to 25 per cent of fixed oils, generally acid-less tallow. In about 90 per cent of the oils, the amount of fixed oil varied from 2 to 7 per cent, and this is to be taken as average. An oil with a flash test lower than 450° F. is to be considered of inferior quality, and should not be used for steam cylinder lubrication.

One of these oils contained free sulphuric acid, and was condemned as unfit, because the acid would corrode the metal surfaces which the oil was to lubricate.

In another sample, about 15% of rosin oil was present. This oil was also condemned, inasmuch as it reacted distinctly acid due to the presence of rosin acids.

In a recent article by Willis Lawrence entitled "Engine-Cylinder Lubrication"* the writer states that "a high-priced oil is not necessarily the best oil for steam cylinder lubrication, also that the moisture content of the steam, a most imporant factor in selecting a good oil, is too often a matter of mere guesswork." This writer greatly favors the employment of graphite, which should be added to the cylinder oil. He made a number of tests, and states in concluding the article that "the results of the experiments justify the opinion that the action of graphite on steam surfaces is mechanical and positive, producing much better lubrication than can be obtained by the use of cylinder oil alone. Also, that no injurious action results from its use either in engine cylinders or the boilers, and, further, that its use increases the efficiency of cylinder oil from 40 to 100 per cent. That deposits will not form any more rapidly than when cylinder oil containing animal fat is used. In most cases, the graphite can be substituted

* Power. Vol. 37, 1913, No. 23.

for the animal fat, when a constant mixture of it and cylinder oil in pre-determined proportions is fed to the cylinder."

There is no doubt but that the addition of finely powdered graphite to lubricating oils is a method of lubrication as yet in its infancy. It represents an upward step in solving the question of minimizing the loss of power on account of friction, and it may be predicted that in the near future, graphite will be a constituent of many lubricants, on account of its superior property of filling the interstices or rough points of wearing surfaces as well as coating them with a fine film, so that extremely smooth surfaces result, providing the application of the graphite is such as will not give rise to other difficulties.

Engine or Machine Oils.

Under this heading come the oils of lighter body but good viscosity that are employed for the lubrication of bearings, journals, air compressors, dynamos and motors, crank case engines, gas and gasoline engines. As a rule, pure mineral oils are used for these purposes. The individual oil that is to be employed for any of these special purposes must be selected according to the conditions under which the machine operates. Where high temperatures prevail, the oil must have a properly high flash and fire test, and be so viscous that it exercises the necessary lubricating effect. If conditions are such that the lubrication must take place at low temperatures, the oil must have a low cold test besides the desired viscosity.

Data compiled from over 50 engine and machine oils analyzed in the laboratories of the Wahl-Henius Institute, Chicago, during one year's time, are as follows:

	Average	Maximum	Minimum
Specific Gravity	0.892	0.916	0.865
Viscosity at 70° F. (Tagliabue Viscosimeter).	241°	428°	160°
Flash Test	396° F.	419° F.	356° F.
Fire Test	434° F.	473° F.	374° F.
Cold Test	22° F.	45° F.	—2° F.

All of these samples represented pure mineral oils, being free from fixed oils. Their reaction was neutral. The average viscosity is somewhat high, but in a good oil it should be at least 180°.

The flash test in a good engine oil should be 375° to 400° F. If it is less than 300° F., there is the possibility that some of the volatile hydrocarbons will evaporate, causing the oil to become more viscous and causing liability of fire. If an oil has too high a flash test, it is also usually of high viscosity and does not flow quickly, so that the friction is increased.

A low-cold test is very desirable in an engine oil, as it is an indication that none or only a very small amount of the hydrocarbons of the paraffin group is present. Such are undesirable, because paraffin is valueless as a lubricant.

Special Engine Oils.

Engine oils which are commonly used for lubricating bearings, journals, dynamos and motors have already been described. In order to obtain the best results, there are, however, a number of purposes, for which an engine oil of special properties should be employed. Among these are ice machine oils, air compressor oils and automobile oils, which are described below.

Ice Machine Oils.—For the lubrication of refrigerating machine compressors, a pure mineral oil having a flash test of at least 300° F. and a cold test of preferably 0° F. and not higher than 5° F. should be employed. Although the viscosity should be good, this property is of minor consideration compared to the flash test and the cold test.

The following are the results obtained from the analysis of over 60 samples of ice machine oils in the laboratories of the Wahl-Henius Institute, Chicago:

	Average	Maximum	Minimum
Specific Gravity	0.913	0.936	0.878
Viscosity at 70° F. (Tagliabue Viscosimeter)	183°	350°	88°
Flash Test	312° F.	347° F.	194° F.
Fire Test	343° F.	383° F.	257° F.
Cold Test	+0.5° F.	+19.4° F.	less than —5° F.

The reason that an ice machine oil must have a low cold test is, so that if any of the oil enters the expansion coils, it will not congeal and form a coating on the inside of the pipes. Should this occur, the coating will act as an insulator and thereby reduce the cooling surface of the pipes. In this case, it will be necessary

to clean all of the expansion coils by first pumping out all of the ammonia and then blowing steam through the coils and carefully draining them. Hot water may also be circulated through them in order to remove the oil.

A flash test lower than 300° F. is especially undesirable in a hot compression type ice machine. Should the compressor be operated fast, there is the danger that on account of the low flashing point, the oil may partly vaporize during the cycle of compression, and in that case, hydrocarbon gases will be produced, which will ultimately fill the condensers. As these gases cannot be liquefied, they will greatly reduce the cooling surface for the condensation of the ammonia, unless they are blown off from time to time. This is a very unpleasant operation, and involves the loss of a considerable quantity of ammonia.

With reference to ice machine lubrication, mention must also be made regarding the oil employed for lubricating the steam cylinders throughout the plant. If the exhaust steam is condensed, purified and used for the manufacture of ice, the separation of the oil from the exhaust steam and condensed water is an important operation. As pointed out under "Cylinder Oils," the greater the amount of saponifiable oil or fat the steam cylinder oil contains, the more difficult is the separation thereof from the exhaust steam and condensed water. Where the latter is used for ice making, a steam cylinder oil with a minimum amount of fixed oil or fats should therefore be employed.

Air Compressor Oils.—In former times, lard oil, tallow oil or glycerin were frequently used. They, however, possess the disadavantage of easily becoming gummy, thereby increasing instead of decreasing the friction. A pure mineral oil of fairly heavy body and a flash test of at least 400° F. and preferably 450° F. is best suited for this purpose. It should have a low cold test, about 25° F., which is an indication that it contains a minimum amount of hydrocarbons of the paraffin group. Such are undesirable, as they have a tendency towards decomposition, especially if the compressor runs hot.

Automobile Lubrication.—The bearings, crank pins and moving parts, excepting the explosion cylinder on motor cars, are usually lubricated according to the splash system. For this purpose, a high grade engine oil with a low cold test is employed.

The oil for lubricating the explosion cylinder must be of such a character that it retains its lubricating effect under the high heat generated in the cylinder by the ignition and explosion of the gasoline vapors. It should be a pure mineral oil, have a flash test of about 350° F., viscosity about 225° (Tagliabue viscosimeter), and a cold test as near 0° F. as possible. If the flash test is too low, the oil will burn and not exercise a lubricating action, and if too high, the viscosity also will be too great, so that the oil does not flow properly. The viscosity must not be too low, as otherwise the oil flows too easily. A low cold test indicates that the oil is practically free from paraffin hydrocarbons, which have a tendency to give rise to carbon formation in the motor cylinder.

Such carbon formation is highly detrimental in gas and gasoline motor operation, and usually gives rise to engine troubles. Frequently the oil used is blamed, but often other conditions are at fault. Principal among these is a faulty mixture of air and gasoline vapors, the carbon being formed when too much gasoline is mixed with insufficient air. In consequence, the carbon contained in the gasoline is not properly burned, and leaves a deposit of soot in the combustion chamber. When this deposit is then intermixed with lubricating oil, especially when it contains appreciable quantities of paraffin group hydrocarbons, it usually forms the hard carbon scale. The feeding of too much lubricating oil also gives rise to such trouble.

Anti-Friction Metals.

The composition of various types of anti-friction metals and their uses have been referred to in other chapters of this book. It is a well-known fact that if the surfaces of two substances rub together the wear on both will be practically equal. If, however, the materials are of different hardness, the softer one will wear more. This principle is applied to bearings and journals of machines, in order to minimize the friction of the wearing surfaces, and that the more costly part of the machine subjected to wear and tear need not be replaced. In other words, the softer material gradually wears and in time will have to be replaced.

Anti-friction metals are to be divided into two classes. The first are the so-called *hard metals,* which are alloys of brass or bronze, and the second are *soft metals,* which are alloys of tin,

lead, copper, antimony and zinc in varying amounts. The hard anti-friction metals are usually applied in the form of a bush, viz., a lining which is inserted in a plate shell clamp or other part of the machinery, so as to receive the wear. The soft metals, which are known as babbitt-metal, white metal, and anti-friction metal, are most commonly used, mainly on account of their cheapness. The percentages of the different metals in the alloy vary according to the pressure and the speed applied on the bearings.

According to W. M. Davis' "Friction and Lubrication," tin imparts toughness and hardness to an alloy and has the property of giving other metals a fine grain. Lead has great anti-frictional value, and gives to the wearing surfaces, a soft, smooth, yielding nature, but of itself will not sustain heavy pressures. Antimony tends to make the alloy hard and brittle, but coarser in grain. Zinc tends to increase the solidity and combine closer the other alloys of copper, tin and lead. Copper gives strength and toughness to the other parts. Bismuth increases the fluidity of the other metals when melted.

Mensuration

Mensuration treats of the measurement of lines, surfaces and volumes.

A *point* is that which has only position, but no magnitude.

A *line* is that which has only one dimension—that is, length, but neither breadth nor thickness. The lines we draw on paper are only symbols to represent the ideal lines.

A straight or *right* line is a continuous line, pursuing the same direction at all points.

A *curved* line is a line of which no portion is straight.

Parallel lines are lines having the same direction and an equal distance from each other at all points.

Figure 255.
Curved Line.

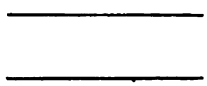


Figure 256.
Parallel Lines.

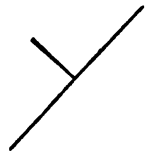


Figure 257.
Perpendicular Line.

A *horizontal* line is a line parallel to the water level or to the horizon.

A *perpendicular* line is a straight line meeting another straight line so as to incline no more to the one side than to the other.

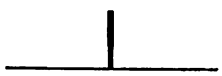


Figure 258. Vertical Line.

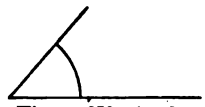


Figure 259. Angle.

Figure 260. Right Angle.

A *vertical* line is a straight line perpendicular to a horizontal line, or a line pointing to the center of the earth.

An *angle* is the difference in direction of two lines proceeding from one common point.

A *right* angle is formed by two lines perpendicular to each other.

An *acute* angle is an angle less than a right angle.

An *obtuse* angle is an angle greater than a right angle.

A *surface* is that which has two dimensions, i. e., length and breadth.

A *plane* is a surface in which a straight line joining any two points of it will lie wholly.

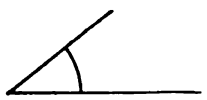


Figure 261. Acute Angle.

Figure 262. Obtuse Angle.

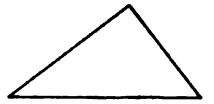


Figure 263. Triangle.

A *plane figure* is a part of a plane surface bounded by straight or curved lines.

The *area* of a plane figure is the surface included within the lines which bound it.

A *triangle* is a plane figure bounded by three sides, composed of straight lines, and having three angles.

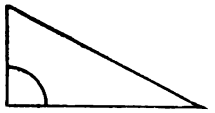


Figure 264.
Right-angled Triangle.

Figure 265.
Obtuse-Angled Triangle.

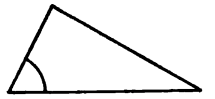


Figure 266.
Acute-Angled Triangle.

A *right-angled* triangle has one right angle.

An *obtuse-angled* triangle has an obtuse angle.

An *acute-angled* triangle has all its angles acute.

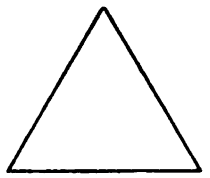


Figure 267.
Equilateral Triangle.

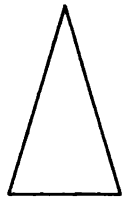


Figure 268.
Isosceles Triangle.

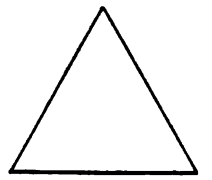


Figure 269.
Base of Triangle.

An *equilateral* triangle has all three sides of equal length.

An *isosceles* triangle has only two sides of equal length.

The *base* of a triangle is the side on which it is supposed to rest.

The *altitude* or *height* of a triangle is a straight line drawn perpendicular to the base from the angle opposite.

The *hypotenuse* is the side opposite the right angle in a right-angled triangle.

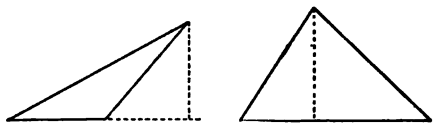


Figure 270.
Altitude or Height of Triangle.

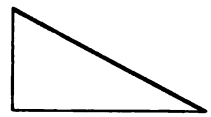


Figure 271.
Hypotenuse.

The *cathetæ* or *legs* are the two sides enclosing the right angle in a right-angled triangle.

Mensuration of Areas.

To find the area of a triangle, multiply the base by the height and divide the product by 2 (see Figure 273).

Area = base × height ÷ 2; *Height* = 2 × area ÷ base; *Base* = 2 × area ÷ height.

Example.—Area = 6 × 10 ÷ 2 = 30.

Height = 2 × 30 ÷ 6 = 10.

Base = 2 × 30 ÷ 10 = 6.

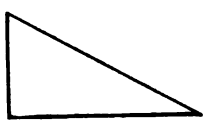


Figure 272.
Cathetae.

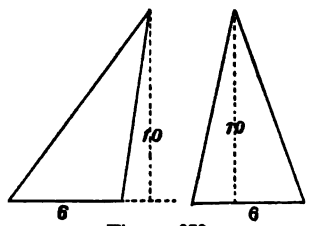


Figure 273.
To Find Area of Triangle.

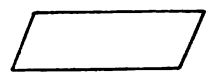


Figure 274.
Parallelogram.

Polygons are plane figures, having three or more sides. They are regular, or irregular, according to whether their sides are of equal length or not; and are named from the number of their sides or angles. A *triangle* is a polygon of three sides or angles. A *quadrilateral* is a polygon of four sides or angles.

Quadrilaterals are divided as follows:

Parallelogram, which is bounded by two pairs of parallel sides.

Trapezoid, having two sides parallel.

Trapezium, having no two sides parallel.

A parallelogram with right angles is called a *rectangle.*

A rectangle whose sides are all equal is called a *square.*

A *pentagon* has five sides. A *hexagon* has six sides. A *heptagon* has seven sides. An *octagon* has eight sides.

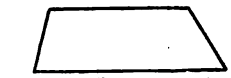


Figure 275. Trapezoid.

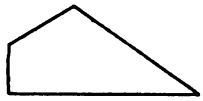


Figure 276. Trapezium.

Figure 277. Rectangle.

The *perimeter* is the boundary line or circumference of a plane figure.

To find the area of a parallelogram, multiply the base by the height, which is the perpendicular distance of the base from the parallel side opposite. Area = ad × h (see Figure 283).

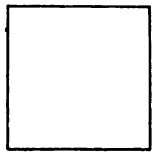


Figure 278. Square.

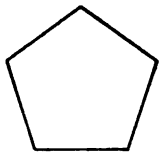


Figure 279. Pentagon.

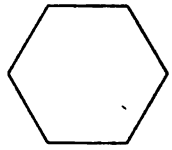


Figure 280. Hexagon.

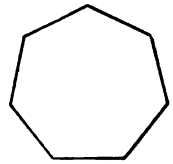


Figure 281. Heptagon.

In a rectangle, the height is equal to the side which is perpendicular to the base. Hence, *to find the area of a rectangle,* multiply the length by the breadth or height. Area = ad × ab = ab × bc (see Figure 284).

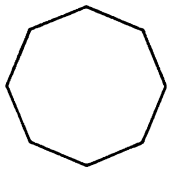


Figure 282. Octagon.

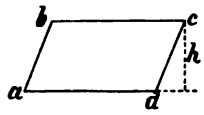


Figure 283. To Find Area of Parallelogram.

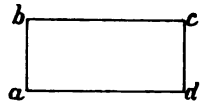


Figure 284. To Find Area of Rectangle.

To find the area of a square, multiply the side by itself. Area = $ad^2 = ab^2 = bc^2 = cd^2$ (see Figure 285).

To find the area of a trapezoid, multiply the sum of the two parallel sides by the height or perpendicular distance between them, and divide the product by 2. Area = (ad + bc) × h ÷ 2 (see Figure 286).

To find the area of a trapezium, divide it into triangles by drawing a diagonal, which is a line connecting two points of the figure not connected by a single side. Multiply the diagonal by the sum of the two perpendiculars falling upon it from the opposite angles, and divide the product by 2. Area = ac (be + df) ÷ 2 (see Figure 287).

To find the area of an irregular polygon, divide the polygon into triangles and add the areas of the triangles. Area = A + B + C (see Figure 288).

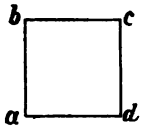


Figure 285. To Find Area of Square.

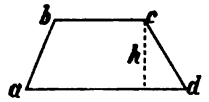


Figure 286. To Find Area of Trapezoid.

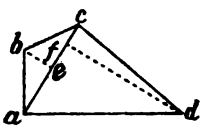


Figure 287. To Find Area of Trapezium.

To find the area of a regular polygon, multiply the length of a side by the perpendicular distance to the center, divide the product by 2, and multiply the quotient by the number of sides (see Figure 289).

$$\text{Area} = \frac{\text{ad} \times \text{oc}}{2} \times 6.$$

Round Figures—Circle.

The *circle* is a plane figure bounded by a curved line, of which all points are at an equal distance from a point within called the *center.*

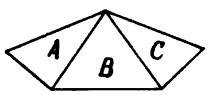


Figure 288. To Find Area of Irregular Polygon.

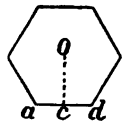


Figure 289. To Find Area of Regular Polygon.

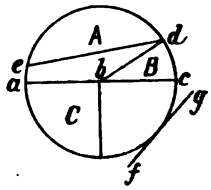


Figure 290. Circle and Parts.

Circumference or *periphery* is the curved line which bounds the circle.

Diameter is a straight line passing through the center and intersecting the circumference on both sides, as abc (see Figure 290).

Radius is a straight line, extending from the center to any point on the circumference, and is one-half the diameter, as bd.

An *arc* of a circle is any part of its circumference, as cd.

A *chord* is any straight line joining two points of the circumference, as ed.

A *segment* is any part bounded by an arc and its chord, as A.

A *sector* is any part of a circle bounded by an arc and its two radii, as B.

A *semicircle* is half a circle.

A *quadrant* is a quarter of a circle, as C.

A *tangent* is a straight line which touches the circle without intersecting it, as fg.

Circumference of a circle = diameter × 3.1416.

Diameter of a circle = circumference ÷ 3.1416.

Area of a circle = square of diameter × 0.7854; or = square of circumference × 0.07958; or = ½ diameter × ½ circumference; or = square of radius × 3.1416.

Examples.—Diameter is 8 feet, what is circumference?

$8 \times 3.1416 = 25.133$ feet.

Circumference is 28 feet, what is diameter?

$28 \div 3.1416 = 8.91$ feet.

Diameter is 8 feet, what is area?

$8 \times 8 \times 0.7854 = 50.266$ square feet.

Circumference is 28 feet, what is area?

$28 \times 28 \times 0.07958 = 62.391$ square feet.

½ diameter is 4 feet, ½ circumference 12.566, what is area?

$4 \times 12.566 = 50.266$ square feet.

Radius = 4, then $4 \times 4 \times 3.1416 = 50.266$ square feet.

Solids.

A solid is that which has three dimensions: Length, breadth and thickness.

A *prism* is any solid whose two ends are parallel, similar and equal, and whose sides are parallelograms. Prisms are triangular, quadrangular, etc., according as the ends are triangles, quadrangles, etc. (see Figures 291-298).

A *right* prism is one whose sides are perpendicular to its ends (see Figures 291, 293, 295, 297, 298).

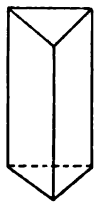


Figure 291. Right Triangular Prism.

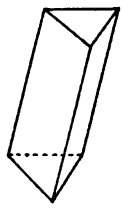


Figure 292. Oblique Triangular Prism.

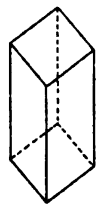


Figure 293. Right Rectangular Prism.

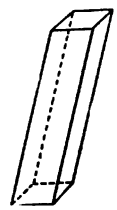


Figure 294. Oblique. Rectangular Prism.

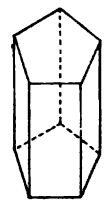


Figure 295. Right Pentagonal Prism.

An *oblique* prism is one whose sides are not perpendicular to its ends (see Figures 292, 294, 296, 297).

When all the sides of the figures which form the ends are equal, and the angles included between those sides are also equal, the prism is said to be *regular* (see Figures 292, 292, 293, 294, 297, 298), otherwise *irregular* (see Figures 295, 296).

A *parallelopiped* is any solid contained within six sides, all of which are parallelograms, and those of each opposite pair, parallel to each other. Parallelopipeds are types of prisms (see Figures 297, 298).

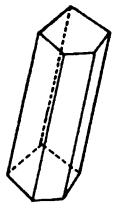
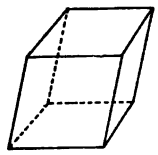
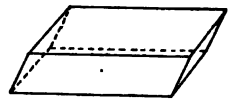


Figure 296. Oblique Pentagonal Prism.





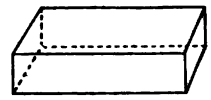


Figure 297. Parallelopipeds.

A *cube* is a right, rectangular prism, or a parallelopiped, having all its sides equal squares, and all its angles right angles. (see Figure 298).

A *cylinder* is any solid whose ends are parallel, similar and equal curved figures, and whose sections, parallel to the ends, are everywhere similar and equal to the ends (see Figures 299-301).

A *right* cylinder is one whose ends are perpendicular to its sides (see Figures 299, 300); when otherwise, it is *oblique* (see Figure 301). The most common form is the *right circular* cylin-

der, whose ends are circles (see Figure 299). Another form frequently used is the *right elliptical* cylinder, whose ends are ellipses (see Figure 300).

The altitude, or height, of a cylinder is the perpendicular distance between the ends.

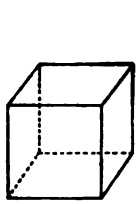


Figure 298. Cube.

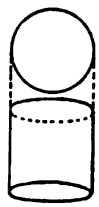


Figure 299. Right Round Cylinder.

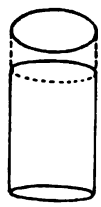


Figure 300. Right Elliptical Cylinder.

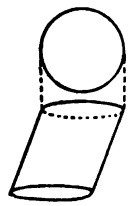


Figure 301. Oblique Round Cylinder.

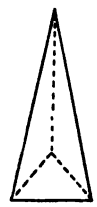


Figure 302. Right Triangular Pyramid.

A *pyramid* is any solid which has for its base a plane figure of any number of sides, and for its sides plane triangles meeting in one point called *vertex* (see Figures 302-305). They are triangular, quadrangular, rectangular, etc., according as the base is

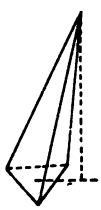


Figure 303. Oblique Triangular Pyramid.

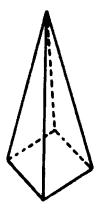


Figure 304. Quadrangular Pyramid.

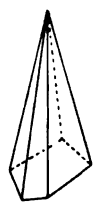


Figure 305. Pentagonal Pyramid.

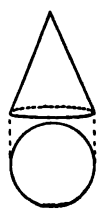


Figure 306. Right Circular Cone.

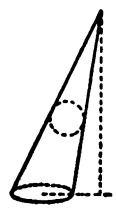


Figure 307. Oblique Circular Cone.

a triangle, quadrangle, rectangle, etc. When the base is a regular figure, the pyramid is said to be *regular* (see Figures 302 and 304), otherwise *irregular* (see Figures 303 and 305). The *altitude* or *height* of a pyramid is the perpendicular distance from vertex to base (see Figure 303).

A *cone* is a solid, of which the base is a curved figure, most commonly a circle, from which the surface tapers uniformly to

a point called *vertex* (see Figures 306 and 307). The *altitude* or *height* of a cone is the perpendicular distance from vertex to base (see Figure 307). Any section of a cone parallel to the base gives a figure similar to the base, but smaller.

A section of a cone or cylinder when cut obliquely by a plane passing through the curved surface, or *mantle,* without intersecting the base, is called an *ellipse* (see Figure 310); o, center; e and f, foci; ac, long diameter or axis; bd, short diameter or axis.

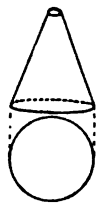


Figure 308.
Frustum of Cone.

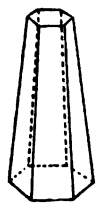


Figure 309.
Frustum of Pyramid.

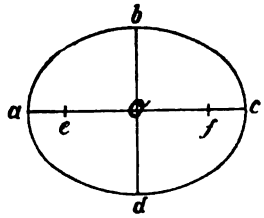


Figure 310.
Ellipse.

To find the area of an ellipse, multiply the product of its semi-axes by 3.1416; or, multiply the product of its axes by 0.7854.

Example.—The long axis is 12 feet, the short axis 8 feet, what is the area?

$6 \times 4 \times 3.1416 = 75.3984$ sq. ft.; or, $12 \times 8 \times 0.7854 =$ 75.3984 sq. ft.

To find the circumference of an ellipse, approximately, D and d being the two axes, use the following formula:

$$\text{Circumference} = 3.1416\sqrt{\frac{D^2 + d^2}{2}}.$$

The *frustum* of a cone or of a pyramid (also called a truncated cone or pyramid) is that part which remains after cutting off the upper part by a plane parallel to the base (see Figures 308 and 309).

A *sphere* is a solid in which every point of the surface is at an equal distance from a point called the center. It is generated by the revolution of a circle around its diameter. The surface of a sphere $= 12.5664\ r^2 = 3.1416$ diameter$^2 = 0.3183$ circumference$^2 =$ diameter $\times$ circumference. The volume of a sphere

is 4.1888 r^3 = 0.5236 $diameter^3$ = 0.01689 $circumference^3$ = 1/6 diameter × area of surface.

Mensuration of Surfaces.

The unit of measure for surface is the *square foot.*

To *find the surface of a right prism,* ascertain the areas of both ends and all sides, and add them together.

Areas of ends + areas of sides.

Example.—The side of the end of a square prism (a rectangular box or bin) is 6 feet, length 15 feet. What is the surface?

6 × 6 =36 = area of one end, × 2 = 72 = area of both ends;
6 × 15 = 90 = area of one side, × 4 = 360 = area of four sides;
72 + 360 = 432 square feet.

To find the surface of a cube (a box of equal length, width and height), ascertain area of square forming its sides, and multiply by 6.

Area of end × 6.

Example.—Side of square is 4 feet. What is the surface of the cube?

4 × 4 = 16 = area of square; 16 × 6 = 96 square feet.

To find the surface of a cylinder, multiply the circumference of the end by the height, and add the areas of both ends.

Height × circumference + 2 × area of end.

Example.—Diameter of round cylinder is 18 inches, the height 72 inches. Find the surface.

18 × 3.1416 = 56.55 = circumference; 56.55 × 72 = 4,071.60.

Area of end is 18^2 × 0.7854 = 254.47; × 2 = 508.94 = area of both ends.

4,071.60 + 508.94 = 4,580.54 square inches.

When internal surface only is to be computed, omit adding areas of top end, or both ends, as the case may be.

The height in an upright cylinder (round tank) is the length in a horizontal cylinder.

To find the surface of a right pyramid, multiply the perimeter of the base by the slant height, measured along the slanting surface (that is, from the vertex to a point midway between two

successive corners of the base), divide the product by two, and add to the quotient the area of the base.

(Perimeter of base $\times$ slant height) $\div 2 +$ area of base.

Example.—In a quadrangular pyramid a side is 8 feet, slant height 24 feet. What is the surface?

$8 \times 4 = 32 =$ perimeter of base; $(32 \times 24) \div 2 = 384 =$ area of sides.

$8 \times 8 = 64 =$ area of base; $384 + 64 = 448$ square feet.

To find the surface of a frustum of a pyramid, add the perimeter of both ends, multiply the sum by the slant height (that is, from a point midway between two successive corners of the base and a point midway between the two corresponding corners of the top), divide the product by 2, and add the areas of both ends.

[(Perimeter of base + perimeter of top) $\times$ slant height] $\div 2 +$ area of base + area of top.

Example.—In a frustum of a quadrangular pyramid the side of the base is 8 feet, of the top 6 feet, slant height 20 feet. What is the surface?

$8 \times 4 = 32$; $6 \times 4 = 24$; $32 + 24 = 56 =$ sum of perimeters;

$56 \times 20 = 1{,}120$, $\div 2 = 560 =$ area of sides;

$8 \times 8 = 64$, area of base; $6 \times 6 = 36 =$ area of top;

$560 + 64 + 36 = 660$ square feet.

To find the surface of a right cone, multiply the circumference of the base by the slant height (that is, from the vertex to the circumference of the base), divide the product by 2 and add the area of the base.

(Circumference of base $\times$ slant height) $\div 2 +$ area of base.

Example.—The diameter of the base is 5 feet, the slant height 15 feet. What is the surface?

$5 \times 3.1416 = 15.708 =$ circumference of base; $15.708 \times 15 = 235.62$, $\div 2 = 117.81$;

$5^2 \times 0.7854 = 19.635 =$ area of base; $117.81 + 19.635 = 137.445$ square feet.

To find the surface of a frustum of a cone, add the circumferences of the two ends, multiply the sum by the slant height (that is, from a point in the circumference of the base to a corresponding point in the circumference of the top, the line so

drawn to lie in a plane perpendicular to the base), divide the product by 2, and add the areas of the two ends.

[(Circumf. of base + circumf. of top) × slant height] ÷ 2 + area of base + area of top.

Example.—The diameter of the base of a frustum of a cone is 10 feet, of the top 8 feet, slant height 12 feet. What is the surface?

$10 \times 3.1416 = 31.416$ = circumference of base;

$8 \times 3.1416 = 25.133$ = circumference of top;

$(31.416 + 25.133) \times 12 = 678.588$, $\div 2 = 339.294$;

$10^2 \times 0.7854 = 78.54$ = area of base.

$8^2 \times 0.7854 = 50.266$ = area of top.

$339.294 + 78.54 + 50.266 = 468.100$ square feet.

To find the surface of the segment of a sphere, the following formula may be used:

$$S = 3.1416 \times h \times \left(\frac{d^2}{4h} + h\right)$$

S = surface of segment of sphere.

h = height of segment.

d = diameter of segment.

Mensuration of Volumes.

The unit of measure of capacity is the cubic foot.

To find the volume of a prism or parallelopiped, multiply the length by the breadth and the depth, or, the area of the base by the height (length).

Length × breadth × depth.

Example.—The three dimensions of a straight box are: Length 6 ft., breadth 2 ft., depth 4 ft. What is the volume?

$6 \times 2 \times 4 = 48$ cubic feet.

To find the volume of a cube, such as a box, bin, etc., where length, breadth and height are equal.

The volume is the cube of the dimension; that is, multiply the dimension twice by itself.

Example.—The side of a cube is 24 inches. What is the volume?

24^3, or $24 \times 24 \times 24 = 13{,}824$ cu. in., or $13{,}824 \div 1{,}728 =$ 8 cu. ft.

To find the size of a box or bin to accommodate a required capacity:

First, reduce bushels to cubic feet; then, if a bin of equal length, width and height (that is, a cube) is wanted, extract the cube root of the number of cubic feet. The result is the inside dimensions of the box.

Example.—Required capacity, 500 bu., dimensions of bin to be equal.

500 bu. = 625 cu. ft.; $\sqrt[3]{625} = 8.55 = 8$ ft. 6½ in.

If the bin is to be square, but not a cube, one of the dimensions must be given.

Example.—Required capacity, 500 bu.; depth, 6 ft. 3 in.

500 bu. = 625 cu. ft.; $625 \div 6.25 = 100$; $\sqrt{100} = 10$. The length and width of the bin are 10 feet.

Example.—Required capacity, 500 bu.; side, 10 ft.

500 bu. = 625 cu. ft.; $625 \div (10 \times 10) = 6.25 = 6$ ft. 3 in.

If the bin is to be simply rectangular, but neither cubic nor square, two of the dimensions must be given.

Example.—Required capacity, 500 bu.; length 12 ft.; depth 6 feet.

500 bu. = 625 cu. ft.; $625 \div (12 \times 6) = 8.68 = 8$ ft. $8^1/_8$ in.

To find the volume of a pyramid or cone, multiply the area of the base by one-third the perpendicular height.

Area of base $\times$ $^1/_3$ height.

Example.—The base of a rectangular pyramid is 3 feet by 4 feet, the height 6 feet. What is the volume?

$$3 \times 4 = 12; 12 \times \frac{6}{3} = 24, \text{ or } \frac{3 \times 4 \times 6}{3} = 24 \text{ cu. ft.}$$

Example.—The diameter of the base of a cone is 3 feet, height 12 feet. What is the volume?

$$\text{Area of base} = 7.069; \frac{7.069 \times 12}{3} = 28.276 \text{ cu. ft.}$$

To find the volume of a cylinder, such as most water tanks, pipes, etc.:

Multiply the area of the base or end by the perpendicular height (or length, in case of a horizontal cylinder).

Area of base $\times$ height.

Example.—The diameter of a round cylinder is 3 feet, the height 12 feet. What is the volume?

Area of circle of 3 feet diameter $= 7.069$; $7.069 \times 12 =$ 84.828 cubic feet.

To find the capacity of a cylinder in gallons, multiply area of bottom in square inches by length in inches, and divide by 231.

Example.—The diameters of an elliptical horizontal cylinder are 8 feet and 6 feet, the length 12 feet. What is the capacity?

$$\frac{96}{2} \times \frac{72}{2} = 48 \times 36 = 1{,}728, \times 3.1416 = 5{,}427.68 \text{ sq. in.}$$

$$5{,}428.68 \times 144 = 781{,}730 \text{ cu. in.}$$

$$781{,}730 \div 231 = 3{,}384.1 \text{ gal.}$$

To find the volume of the segment of a sphere, the following formula may be used:

$$V = [(r^2 \times 3) + h^2] \times h \times 0.5236$$

V = volume of segment.

r = radius of segment.

h = height of segment.

Capacities of Tanks, Tubs, Cisterns, Casks, Bins, Etc.

The formulas for rectangular prisms, parallelopipeds, cubes, pyramids, cones, cylinders, frustums of pyramids or cones, apply to all the boxes and vessels of regular geometrical shape used in the plant. For those of irregular shape, approximate working formulas are given.

For calculating capacities, take inside measurements.

Formula for calculating capacity of a round tank: (Diameter in feet)$^2 \times$ depth in feet $\times 0.7854 =$ cubic feet; (diameter in feet)$^2 \times$ depth in feet $\times 5.878 =$ gallons; (diameter in feet)$^2 \times$ depth in feet $\times 0.1865 =$ barrels of 31½ gallons. Cubic feet $\div$ 4.144 = barrels of 31 gallons.

Example.—Diameter of tank 6 feet, depth 10 feet. What is the volume?

$36 \times 10 \times 0.7854 = 282.74$ cubic feet.

$36 \times 10 \times 5.878 = 2{,}116.08$ gallons.

$36 \times 10 \times 0.1865 = 67.14$ barrels of 31½ gallons.

$282.74 \div 4.144 = 68.23$ barrels of 31 gallons.

To find the volume of a frustum of a pyramid or cone: This is the shape of most tubs and tanks used in the fermenting cellar and storage cellar of a brewery.

1. 1/3 perpendicular height $\times$ (area of top + area of base + $\sqrt{\text{area of top} \times \text{area of base}}$).

2. For frustum of conical vessels only (approximate value):

$$\left(\frac{\text{Top diam.}^2 + \text{bottom diam.}^2}{2}\right) \times 0.7854 \times \text{height.}$$

or,

$$\frac{\text{Top area} + \text{bottom area}}{2} \times \text{height.}$$

Example.—The inside measurements of a fermenting tub are: Bottom diameter, 10 feet; top diameter, 9.5 feet; height, 7 feet. What is the capacity?

1. $\frac{1}{3} \times 7 \times (78.54 + 70.88 + \sqrt{78.54 \times 70.88}) = \frac{1}{3} \times (149.42 + 74.61) = 522.74$ cu. ft.
2. $\left(\frac{9.5^2 + 10^2}{2}\right) \times 0.7854 \times 7 = 522.97$ cu. ft.
3. $\frac{70.88 + 78.54}{2} \times 7 = 522.97$ cu. ft.

Abridged working formula for calculating the capacity of round tanks in barrels of 31 gallons:

1. Reduce all measurements to inches; add diameters of bottom and top, and divide by 2, to obtain mean diameter. Square the mean diameter, multiply this by 0.00011, and the product by the height.

Example.—Bottom diameter, 10 feet; top diameter, 9.5 feet; height, 7 feet.

$\frac{120 + 114}{2} = 117$; $117^2 = 13{,}689$, $\times$ 0.00011 = 1.506, $\times$ 84 = 126.50 barrels.

2. Reduce all measurements to feet; find mean diameter as above. Square the mean diameter, multiply this by 0.19, and the product by the height.

Example.—Same dimensions as above.

$\frac{10 + 9.5}{2} = 9.75$; $9.75^2 = 95.06$, $\times$ 0.19 = 18.06, $\times$ 7 = 126.42 barrels.

Manufacturers of tanks commonly calculate the capacities of tanks in the following way, the taper of the tanks being so slight —about ¾ inch to the rising foot—that the difference between the height of the vessel and the length of the side is ignored. Square the mean diameter of the tank in inches, multiply this by the length of the stave in inches, and the product by 0.0034; the

result is the capacity in gallons. To obtain the length of the stave for this purpose, take off 2 inches from the actual extreme length, then take off 5 inches each for top and bottom for 2-inch lumber, and 6 inches each for 3-inch lumber.

(Mean diameter)^2 $\times$ stave $\times$ 0.0034 = gal.; stave = actual length $-$ 2 in. $-$ 2 $\times$ 5 (or 2 $\times$ 6) in.

Example.—Mean diameter 12 ft., full length of stave 16 ft., 2-inch lumber. Find the capacity.

Length of stave 16 feet, less 2 inches, less 2 $\times$ 5 inches = 15 feet. Hence,

144^2 $\times$ 180 $\times$ 0.0034 = 12,690.432 gals.=409.369 beer barrels.

United States Measurements of Cisterns and Tanks.—The above agrees substantially with the rule given to gaugers of the United States internal revenue service for use where the inside measurements of a cistern cannot be taken, as follows: "Take the outside circumference of the cistern, half way between the bottom and top; divide this by 3.1416 (or multiply by 7 and divide by 22), and you have the mean diameter on the outside; deduct from this twice the thickness of the staves, and you have the mean inside diameter. Multiply this sum by itself and by the height and the product by 0.0034. The product is the capacity of the cistern in gallons."

To find the capacity of a cask (approximately):

1. Reduce all measurements to inches. Square the mean diameter, multiply this by length, and the product by 0.0034. The mean diameter is one-half the sum of the head and bung diameters, or, according to some authorities, the sum of one-third of the head diameter and two-thirds of the bung diameter. The result is the capacity in gallons.

(Mean diameter)^2 $\times$ length $\times$ 0.0034.

Example.—Bung diameter 24 inches, head diameter 16 inches, length 36 inches.

$$\text{Mean diameter} = \frac{24 + 16}{2} = 20; 20 \times 20 \times 36 \times 0.0034 =$$

48.96 gals.

2. Reduce all measurements to inches.

$$\left(\frac{\text{Head diam.} + \text{bung diam.}}{4}\right)^2 \times 3.14 \times \text{length}.$$ This gives the capacity in cubic inches.

Example: $\left(\frac{16+24}{4}\right)^2 \times 3.14 \times 36 = 11{,}304$ cu. in.

United States Measurement of Casks.—The regulations of the United States internal revenue office classify casks into three varieties, the distinctions being based on the curvature of the staves at the quarter hoops, that is, midway between the bung and chimb. Following are the rules:

Variety 1 (least curvature).—Multiply difference between head diameter and bung diameter by 0.55, and add to product the head diameter. This gives the mean diameter.

Variety 2 (medium curvature).—Same, with decimal 0.63.

Variety 3 (greatest curvature).—Same, with decimal 0.70.

Then multiply the square of the mean diameter, in inches, by 0.0034 and by length in inches. The product is the capacity in gallons. Thus, for the dimensions 24, 16, 36, the capacities for the different varieties are (1) 50.94, (2) 54.18, (3) 57.1.

The familiar fact that casks seldom come up to their nominal capacity is due in part to a slight inward curvature of the heads, for which no allowance is anywhere provided.

Ullage is the amount of liquor in a cask when not full.

To find the ullage of a cask:

1. For a lying cask: Divide the number of wet or dry inches by the bung diameter in inches; if the quotient is less than 0.5, deduct from it one-fourth part of what it is short of 0.5; if it exceeds 0.5 add to it one-fourth of the excess over 0.5; multiply the remainder or sum by the contents of the whole cask in gallons. The product is the quantity of liquor in the cask in gallons, when the dividend is wet inches, or the empty space if dry inches.

2. For a standing cask: Divide the number of wet or dry inches by the length of the cask in inches. If the quotient exceeds 0.5, add to it one-tenth of the excess above 0.5; if less than 0.5, subtract from it one-tenth of what it is short of 0.5; multiply the sum or the remainder by the contents of the whole cask in gallons. The product is the quantity of liquor in the cask, if the dividend is wet inches, or the empty space if dry inches.

To find the volume of a combined square box and pyramid; that is, the capacity of a *bin with hopper* and vessels of similar shape. This working formula is close enough for practical purposes, although not mathematically true, as it makes no allowance

for the slight inaccuracy resulting from the fact that the inverted pyramid is not perfect, but rather a frustum, the vertex being cut off by the trap at the bottom.

Example.—In a bin with hopper the box is 5 feet long, 5 feet wide, 4 feet 9 inches deep. The hopper (pyramid) is 2 feet 6 inches deep. What is the capacity?

Vol of box $= 5 \times 5 \times 4.75$ $= 118.75$ cu. ft.

Vol. of pyramid $= \frac{5 \times 5 \times 2.50}{3}$ $= 20.833$ cu. ft.

Capacity of bin and hopper $= 139.583$ cu. ft. $= 112.1$ bu.

To find the volume of a combined round cylinder and cone; that is, the capacity of a *steep tank with hopper* and similar vessels.

Add the volumes of the cylinder and of the cone.

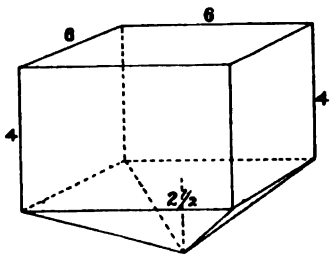
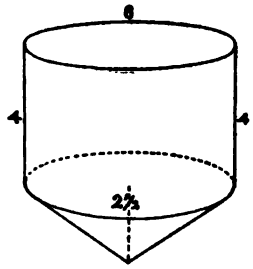


Figure 311.
Square Box with Pramid.



Firgure 312
Round Cylinder with Cone.

Example.—In a steep tank, the tank proper (cylinder) measures 6 ft. diameter and 4 ft. depth, the hopper (inverted cone) 2 ft. depth. What is the capacity?

Vol. of cylinder $= 6^2 \times 0.7854 \times 4 =$ 113.0976 cu. ft.

Vol. of cone $= 6^2 \times 0.7854 \times 2/3 =$ 18.8496 cu. ft.

Capacity of steep tank and hopper = 131.9472 cu. ft.
$= 106$ bu.

Weights and Measures

The measures of extension and weight in use in the United States are derived from the British, and, for all practical purposes, are identical with them. The measures of volume, however, both liquid and dry, show marked differences.

Measures of Length (Linear Measures, or Measures of Extension.)

The standard unit of length is the *yard,* measured on a silver scale inlaid in a bronze bar, at 62° F.

Theoretically, this is derived from a pendulum vibrating seconds at sea level in the latitude of London, in vacuum at 62° F. The length of such a pendulum is divided into 39.1393 parts, called *inches,* 36 of which constitute a yard. The British standard having been adopted by the United States Coast and Geodetic Survey, is in use in all the departments of the Federal government and generally throughout the country.

Since 1893 the Office of Weights and Measures has been authorized to derive the yard from the *meter,* using for the purpose the relation legalized in 1866, namely, 1 yard $= \frac{3,600}{3,937}$ meter.

The following are common measures of length.

Measures of Length.

12 inches (in.) = 1 foot (ft.).
3 feet = 1 yard (yd.).
5½ yards or 16½ feet = 1 rod (rd.), pole, or perch.
40 rods = 1 furlong (furl.).
8 furlongs = 1 mile (mi.)

mi.	furl.	rd.	yd.	ft.	in.
1 =	8 =	320 =	1760 =	5280 =	63360
	1 =	40 =	220 =	660 =	7920
		1 =	5½ =	16½ =	198
			1 =	3 =	36
				1 =	12

72 points = 1 inch.
6 points = 1 line.
12 lines = 1 inch.

4 inches = 1 hand.
3 inches = 1 palm.
9 inches = 1 span.

8 spans, or 6 feet, or 2 yards = 1 fathom.

120 fathoms = 720 feet = 1 cable's length.
6080.27 feet = 1.1516 statute miles = 1 geographical or nautical mile (knot*).
3 nautical miles = 1 league.
60 nautical miles (69.09 stat. mi.) = 1 degree { latitude on a meridian, or longitude on the equator.
360 degrees = circumference of the earth at the equator.

Fathoms and cables are used for measuring rope; points and lines for pendulums.

Surveyor's Linear Measure.

7.92 inches = 1 link.
25 links = 1 rod.
66 feet, or 4 rods = 1 Gunter's chain (ch.).
80 chains = 1 mile.

mi.	ch.	rd.	link	in.
1 =	80 =	320 =	8000 =	63360
	1 =	4 =	100 =	792
		1 =	25 =	198
			1 =	7.92

Surface or Square Measure.

144 square inches (sq. in.) = 1 square foot (sq. ft.).
9 square feet = 1 square yard (sq. yd.) = 1,296 square inches.
40 square poles, or rods = 1 rood.
30¼ square yards, or 272¼ square feet = 1 square rod (sq. rd.), or perch (P.).
4 roods, 10 square chains, 160 square rods, 4840 square yards, or 43,560 square feet = 1 acre (A.).
640 acres = 1 square mile (sq. mi.).
1 sq. mi. = 640 acres, =102,400 sq. rds., = 3,097,600 sq. yds., = 27,878,400 sq., ft., = 4,014,489,600 sq. in.
100 square feet = 1 square (Arch. and Build. measure).

Land Measure (Surveyors' Square Measure).

625 square links (sq. l.) = 1 pole (P.).
16 poles = 1 square chain (sq. ch.).
10 square chains = 1 acre.
640 acres = 1 square mile, or section.
160 acres = 1 quarter section.
36 square miles (6 miles square) = 1 township (Tp.).
1 township = 36 square miles = 23,040 acres.

* As generally used, 1 knot is a speed of 1 nautical mile per hour.

Inches in Decimals of a Foot.

Ins.	Feet.	Ins.	Feet.	Ins.	Feet.	Ins.	Feet.	Ins.	Feet.	Ins.	Feet.
0	.0000	2	.1667	4	.3333	6	.5000	8	.6667	10	.8333
1-32	.0026		.1693		.3359		.5026		.6693		.8359
1-16	.0052		.1719		.3385		.5052		.6719		.8385
3-32	.0078		.1745		.3411		.5078		.6745		.8411
1/8	.0104	1/8	.1771	1/8	.3438	1/8	.5104	1/8	.6771	1/8	.8438
5-32	.0130		.1797		.3464		.5130		.6797		.8464
3-16	.0156		.1823		.3490		.5156		.6823		.8490
7-32	.0182		.1849		.3516		.5182		.6849		.8516
1/4	.0208	.1875	.1875	.3542	.3542	.5208	.5208	.6875	.6875	.8542	.8542
9-32	.0234		.1901		.3568		.5234		.6901		.8568
5-16	.0260		.1927		.3594		.5260		.6927		.8594
11-32	.0286		.1953		.3620		.5286		.6953		.8620
3/8	.0312	3/8	.1979	3/8	.3646	3/8	.5312	3/8	.6979	3/8	.8646
13-32	.0339		.2006		.3672		.5339		.7005		.8672
7-16	.0365		.2031		.3698		.5365		.7031		.8698
15-32	.0391		.2057		.3724		.5391		.7057		.8724
1/2	.0417	1/2	.2083	1/2	.3750	1/2	.5417	1/2	.7083	1/2	.8750
17-32	.0443		.2109		.3776		.5443		.7109		.8776
9-16	.0469		.2135		.3802		.5469		.7135		.8802
19-32	.0495		.2161		.3828		.5495		.7161		.8828
5/8	.0521	5/8	.2188	5/8	.3854	5/8	.5521	5/8	.7188	5/8	.8854
21-32	.0547		.2214		.3880		.5547		.7214		.8880
11-16	.0573		.2240		.3906		.5573		.7240		.8906
23-32	.0599		.2266		.3932		.5599		.7266		.8932
3/4	.0625	3/4	.2292	3/4	.3958	3/4	.5625	3/4	.7292	3/4	.8958
25-32	.0651		.2318		.3984		.5651		.7318		.8984
13-16	.0677		.2344		.4010		.5677		.7344		.9010
27-32	.0703		.2370		.4036		.5703		.7370		.9036
7/8	.0729	7/8	.2396	7/8	.4063	7/8	.5729	7/8	.7396	7/8	.9063
29-32	.0755		.2422		.4089		.5755		.7422		.9089
15-16	.0781		.2448		.4115		.5781		.7448		.9115
31-32	.0807		.2474		.4141		.5807		.7474		.9141
1	.0833	2	.2500	5	.4167	7	.5833	9	.7500	11	.9167
1-32	.0859		.2526		.4193		.5859		.7526		.9193
1-16	.0885		.2552		.4219		.5885		.7552		.9219
3-32	.0911		.2578		.4245		.5911		.7578		.9245
1/8	.0938	1/8	.2604	1/8	.4271	1/8	.5938	1/8	.7604	1/8	.9271
5-32	.0964		.2630		.4297		.5964		.7630		.9297
3-16	.0990		.2656		.4323		.5990		.7656		.9323
7-32	.1016		.2682		.4349		.6016		.7682		.9349
1/4	.1042	1/4	.2708	1/4	.4375	1/4	.6042	1/4	.7708	1/4	.9375
9-32	.1068		.2734		.4401		.6068		.7734		.9401
5-16	.1094		.2760		.4427		.6094		.7760		.9427
11-32	.1120		.2786		.4453		.6120		.7786		.9453
3/8	.1146	3/8	.2813	3/8	.4479	3/8	.6146	3/8	.7813	3/8	.9479
13-32	.1172		.2839		.4505		.6172		.7839		.9505
7-16	.1198		.2865		.4531		.6198		.7865		.9531
15-32	.1224		.2891		.4557		.6224		.7891		.9557
1/2	.1250	1/2	.2917	1/2	.4583	1/2	.6250	1/2	.7917	1/2	.9583
17-32	.1276		.2943		.4609		.6276		.7943		.9609
9-16	.1302		.2969		.4635		.6302		.7969		.9635
19-32	.1328		.2995		.4661		.6328		.7995		.9661
5/8	.1354	5/8	.3021	5/8	.4688	5/8	.6354	5/8	.8021	5/8	.9688
21-32	.1380		.3047		.4714		.6380		.8047		.9714
11-16	.1406		.3073		.4740		.6406		.8073		.9740
23-32	.1432		.3099		.4766		.6432		.8099		.9766
3/4	.1458	3/4	.3125	3/4	.4792	3/4	.6458	3/4	.8125	3/4	.9792
25-32	.1484		.3151		.4818		.6484		.8151		.9818
13-16	.1510		.3177		.4844		.6510		.8177		.9844
27-32	.1536		.3203		.4870		.6536		.8203		.9870
7/8	.1563	7/8	.3229	7/8	.4896	7/8	.6563	7/8	.8229	7/8	.9896
29-32	.1589		.3255		.4922		.6589		.8255		.9922
15-16	.1615		.3281		.4948		.6615		.8281		.9948
31-32	.1641		.3307		.4974		.6641		.8307		.9974

Circular Measure.

60 seconds, $'' = 1$ minute, $'$ | 90 degrees = 1 quadrant.
60 minutes, $' = 1$ degree, $^\circ$ | 360 degrees = circumference.

A circular inch is the area of a circle 1 inch in diameter = 0.7854 square inch; 1 square inch = 1.2732 circular inches.

Measures of Weight.

The legal standard of weight for the United States is the *Troy pound* of the Philadelphia mint, and is an exact copy of the Imperial Troy pound of Great Britain, containing 5,760 grains.

The standard in common commercial use in the United States is the *avordupois* or *commercial* pound, which contains 7,000

grains, and is practically identical with the British avordupois pound, which is the weight of 27.7015 cubic inches of distilled water in air at 39.2° F. with a barometer of 30 inches.

While the pound is the theoretical standard, the practical unit is the grain, which is the same in Troy, apothecaries' and avoirdupois or commercial weight.

Avoirdupois or Commercial Weight.

This is the measure used in all ordinary business transactions, and is understood to be taken for the standard, unless otherwise specified.

16 drams = 1 ounce (oz.) = 437.5 grains (gr.).
16 ounces = 1 pound (lb.) = 7,000 grains.
28 pounds = 1 quarter (qr.).
4 quarters = 1 hundredweight (cwt.) = 112 pounds.
20 hundredweights = 1 ton of 2,240 lbs., or long ton.
2000 pounds = 1 net, or short ton.
2204.6 pounds = 1 metric ton.
1 stone = 14 pounds; 1 quintal = 100 pounds.

The long, or gross ton, hundredweight and quarter were formerly in common use, but now are used almost exclusively by the United States custom houses, and in freighting coal and selling it by wholesale.

Troy Weight.

24 grains = 1 pennyweight (pwt., or dwt.).
20 pennyweights = 1 ounce.
12 ounces = 1 pound.

lb.	oz.	pwt.	gr.
1 =	12 =	240 =	5760
	1 =	20 =	480
		1 =	24

Troy weight is used for gold and silver.

A *carat* is used by jewelers in weighing precious stones. In the United States it is equal to 3.2 grains or 0.205 gram. In London it is equal to 3.17 grains, in Paris, 3.18 grains, which is subdivided into 4 *jewelers' grains.*

The term *carat* is also used to designate the fineness of gold, pure gold being 24 carat, and alloyed gold, according to the ratio of gold and alloy, 22, 20, 16, etc., carat.

Apothecaries' Weight.

20 grains = 1 scruple (sc. or ℈).
3 scruples = 1 dram (dr. or ʒ) = 60 grains.

8 drams = 1 ounce (oz. or ℥) = 480 grains.
12 ounces = 1 pound (lb. or ℔.) = 5,760 grains.
℔ = ℥ 12 = ℨ 96 = ℈ 288 = gr. 5,760.

The pound, ounce and grain of apothecaries' weight are the same as those of Troy weight, the ounce being differently divided.

Miscellaneous Weights.

The following weights are in general use, either by virtue of custom or of law:

32 lbs. of oats = 1 bu.
48 lbs. of barley = 1 bu.
56 lbs. of rye = 1 bu.
56 lbs. of ind. corn = 1 bu.
50 lbs. of ind. meal = 1 bu.
60 lbs. of wheat = 1 bu.
100 lbs. of grain or flour = 1 cental.
100 lbs. of meal or flour = 1 sack.
100 lbs. of dry fish = 1 quintal.
100 lbs. of nails = 1 cask.
196 lbs. of flour = 1 barrel.
200 lbs. of beef or pork=1 barrel.

Comparison of Different Weights.

In computing weights given in different measures (Avoirdupois, Troy and Apothecaries') in any of the other measures, reduce all to *grains,* the grain being the same in the three standards.

	Troy.	Avoirdupois.	Apothecaries'.
1 pound =	5,760 grains =	7,000 grains =	5,760 grains.
1 ounce =	480 grains =	437.5 grains =	480 grains.
	175 pounds =	144 pounds =	175 pounds.

Coal Measure.

Anthracite: 1 cu. ft. = 1.75 broken.
Anthracite: 50-55 lbs. = 1 cu. ft.
Anthracite: 41-45 cu. ft. = 1 ton broken.
Bituminous: 70-78 lbs. = 1 heaped bushel.
Bituminous: 40-50 lbs. = 1 cu. ft.
Charcoal (hardwood): 18.5 lbs. = 1 cu. ft.
Charcoal (pinewood): 18 lbs. = 1 cu. ft.

Measures of Capacity or Volume.

The measures of capacity in use in the United States have no exact equivalent in the measures of any other country. While many names are derived from the British, and, in some cases, are identical with them, their values, as a rule, are quite different.

The units of capacity are the *gallon* for liquids, and the *bushel* for dry measure.

The *standard liquid gallon* of the United States contains 231 cubic inches, and is equal to 8.3389 pounds avoirdupois of pure water at 39.2° F. at a barometric pressure of 30 inches. The *half peck* or *dry gallon* contains 268.8 cubic inches. The *imperial gallon* of Great Britain contains 277.410 cubic inches. It was originally taken as the volume occupied by 10 pounds of water weighed in air at 62° F. and 30 inches barometric pressure. It is equal to about 1.2 United States liquid gallons.

The *standard struck bushel* of the United States contains 2150.42 cubic inches (the old British Winchester struck bushel), or 1.2445 cubic feet, or 77.6274 pounds avoirdupois of pure water at 39.2° F. It is a cylindrical measure 18½ inches in diameter and 8 inches deep.

A *heaped bushel,* which is like the foregoing, with a heaped cone not less than 6 inches high, is equal to 1¼ struck bushels.

The *imperial bushel* of Great Britain contains 2219.28 cubic inches, or 1.2843 cubic feet.

The United States *standard barrel* contains 31½ gallons.

The United States *beer barrel,* according to the Internal Revenue laws of the United States, contains 31 gallons even.

Cubic or Solid Measures.

1,728 cubic in. = 1 cubic ft.	cu. yd.		cu. ft.		cu. in.
27 cubic ft. = 1 cubic yd.	1	=	27	=	46,656

Liquid Measure.

	gal.		qt.		pt.		gi.
4 gills (gi.) = 1 pint (pt.).	1	=	4	=	8	=	32
2 pints = 1 quart (qt.).			1	=	2	=	8
4 quarts = 1 gallon (gal.).					1	=	4

At maximum density, 1 cubic foot of water weighs about 62.5 pounds, and 1 gallon equals about 8.34 pounds. One United States gallon equals 231 cubic inches.

To reduce cubic inches to gallons, divide by 231; cu. in. ÷ 231 = gal.

To reduce gallons to cubic inches, multiply by 231; gal. × 231 = cu. in.

To reduce gallons to cubic feet, divide by 7.48. To reduce cubic feet to gallons, multiply by 7.48.

One beer barrel (bbl.) contains 31 gal. = 4.144 cu. ft. = 7,161 cu. in.

United States Gallons in Cubic Feet.

Gallons.	Cubic Ft.	Gallons.	Cubic Ft.	Gallons.	Cubic Ft.
1	.134	1,000	133.681	1,000,000	133,680.6
2	.267	2,000	267.361	2,000,000	267,361.1
3	.401	3,000	401.042	3,000,000	401,041.7
4	.535	4,000	534.722	4,000,000	534,722.2
5	.668	5,000	668.403	5,000,000	668,402.8
6	.802	6,000	802.083	6,000,000	802,083.3
7	.936	7,000	935.764	7,000,000	935,763.9
8	1.069	8,000	1,069.444	8,000,000	1,069,444.4
9	1.203	9,000	1,203.125	9,000,000	1,203,125.0
10	1.337	10,000	1,336.806	10,000,000	1,336,805.6

Cubic Feet in United States Gallons.

Cubic Ft.	Gallons.	Cubic Ft.	Gallons.	Cubic Ft.	Gallons.
0.1	0.75	50	374.0	8,000	59,844.2
0.2	1.50	60	448.8	9,000	67,324.7
0.3	2.24	70	523.6	10,000	74,805.2
0.4	2.99	80	598.4	20,000	149,610.4
0.5	3.74	90	673.2	30,000	224,415.6
0.6	4.49	100	748.0	40,000	299,220.8
0.7	5.24	200	1,496.1	50,000	374,025.9
0.8	5.98	300	2,244.2	60,000	448,831.1
0.9	6.73	400	2,992.2	70,000	523,636.3
1	7.48	500	3,740.3	80,000	598,441.5
2	14.96	600	4,488.3	90,000	673,246.7
3	22.44	700	5,236.4	100,000	748,051.9
4	29.92	800	5,984.4	200,000	1,496,103.8
5	37.40	900	6,732.5	300,000	2,244,155.7
6	44.88	1,000	7,480.5	400,000	2,992,207.6
7	52.36	2,000	14,961.0	500,000	3,740,259.5
8	59.84	3,000	22,441.6	600,000	4,488,311.4
9	67.32	4,000	29,922.1	700,000	5,236,363.3
10	74.80	5,000	37,402.6	800,000	5,984,415.2
20	149.6	6,000	44,883.1	900,000	6,732,467.1
30	224.4	7,000	52,363.6	1,000,000	7,480,519.0
40	299.2				

To reduce cubic feet to beer barrels, divide by 4.144; cu. ft. $\div$ 4.144 = beer bbl.

To reduce beer barrels to cubic feet, multiply by 4.144, beer bbl. $\times$ 4.144 = cu. ft. For practical purposes 4.14 may be used.

One cubic foot = 0.24 beer bbl.; one cubic inch = 0.00014 beer bbl.

Barrels and hogsheads are fixed measures only for the articles on which an Internal Revenue tax is levied, as beer, whisky, etc. For ordinary commercial purposes they are not fixed measures.

A hogshead of beer is two barrels, or not to exceed 63 gallons.

The United States beer barrel, under the Internal Revenue law, as compared with other measures, is as follows:

Beer Barrels.	Bushels.	Cubic Feet.	Gallons.	Cubic Inches.
1.	3.33	4.144	31.	7161.
0.328	1.	1.244	9.309	2150.4
0.2413	0.804	1.	7.476	1728.
0.0323	0.1074	0.1337	1.	231.

Liquid Measure.

	Cubic Inches.	Pounds Avoir.
Gill	7.21875 cu. in.	0.2604375 lbs.
Pint	28.875 "	1.04195 "
Quart	57.750 "	2.0839 "
Gallon	231. "	8.3356 "
Wine Barrel, 31½ gals.	7276.5 "	262.5714 "
Beer Barrel, 31 gals	7161. "	258.4036 "

For Cisterns, Tanks, Reservoirs, Etc.

31½ gal. = 1 barrel.
63 gal. = 1 hogshead (hhd.).

hhd.	bbl.	gal.	qt.	pt.
1 =	2 =	63 =	252 =	504
	1 =	31½ =	126 =	252

The contraction in volume of water in cooling from boiling to greatest density is about 4.5 per cent. To find the volume that boiling hot water will occupy when so cooled, multiply the volume of hot fluid by 95½ and divide by 100, or multiply by 0.955.

United States standard barrel contains 31½ gals. = 4.21 cu. ft. (in practice often figured roughly at 4¼ cu. ft.).

To reduce cubic feet to United States standard barrels, divide by 4.21 (roughly 4.25); cu. ft. ÷ 4.21 = standard bbl.

To reduce United States standard barrels to cubic feet, multiply by 4.21 (roughly 4.25); standard bbl. × 4.21 = cu. ft.

Apothecaries' Fluid Measure.

60 minims, or drops = 1 fluid drachm.
8 fluid drachms = 1 fluid ounce.
16 fluid ounces = 1 pint.
8 pints = 1 gallon.

Miscellaneous Fluid Measures.

Tierce = 42 gals.
Butt of sherry = 108 gals.
Pipe of port = 115 gals.
Pipe of Teneriffe = 100 gals.
Butt of Malaga = 105 gals.
Puncheon of Scotch whisky = 110-130 gals.
Puncheon of brandy = 110-120 gals.
Puncheon of rum = 100-110 gals.
Hogshead of brandy = 55-60 gals.
Hogshead of claret = 46 gals.

A hogshead is ½, a quarter¼, an octave ⅛ of a pipe, butt or puncheon.

To reduce United States liquid measures to British, of the same denomination, divide by 1.2; or to reduce British to United States, multiply by 1.2. The British Imperial gallon contains 277.410 cubic inches, or is equal to about 1.2 United States gallons.

Dry Measure.

One United States bushel = 2,150.42 cu. in. = 1.2445 cu. ft. = 77.6274 pounds avoirdupois of pure water at 39.2° F.; 1 cu. ft. = 0.80356 struck bushel.

To reduce cubic feet to bushels, divide by 1.24; cu. ft. ÷ 1.24 = bu. For practical purposes subtract from cubic feet one-fifth of their number to obtain bushels; cu. ft. — 1-5 cu. ft. = bu.

To reduce bushels to cubic feet, multiply by 1.24; bu. × 1.24 = cu. ft. For practical purposes add to the number of bushels one-fourth of their number to obtain cubic feet; bu. + ¼ bu. = cu. ft.

The half-peck or dry gallon contains 268.8 cubic inches.

	bu.	pk.	qt.	pt.	cu. in.
2 pints (pt.) = 1 quart (qt.)	1	= 4	= 32	= 64	= 2,150.42
8 quarts = 1 peck (pk.).		1	= 8	= 16	= 537.6
4 pecks = 1 bushel (bu.).			1	= 2	= 67.2

A heaped bushel contains 2,747.715 cubic inches.

The dry flour barrel = 3.75 cu. ft. = 3 struck bushels. The dry barrel is not a fixed measure, however, and barrels vary considerably in size.

The imperial bushel of Great Britain contains 2219.28 cubic inches or 1.2843 cubic feet.

To reduce United States dry measures to British, of the same denomination, divide by 1.032; to reduce British into United States measures, multiply by 1.032.

Comparative Table of Measures of Capacity.

	Cu. In. in One Gallon.	Cu. In. in One Quart.	Cu. In. in One Pint.	Cu. In. in One Gill.
Liquid measure	231	57¾	28⅞	7 7/32
Dry measure	268⅘	67⅕	33⅗	8⅖

Shipping Measure.

100 cu. ft. = 1 register ton.

U. S. shipping ton = 32.143 U. S. bu. = 31.145 Imper. bu. = 40 cu. ft.

British shipping ton = 33.75 U. S. bu. = 32.702 Imper. bu. = 42 cu. ft.

Wood Measure.

In measuring wood, a pile of wood cut 4 feet long, piled 4 feet high and 8 feet on the ground, is called a cord.

16 cubic feet = 1 cord foot (cd. ft.).

8 cord feet, or 128 cubic feet = 1 cord (cd.).

In board measure, boards are assumed to be one inch in thickness. To obtain the number of feet board measure (B. M.) of a board, or piece of square timber, multiply length in feet, breadth in feet and thickness in inches.

1 board ft. = 1 ft. long, 1 ft. wide, 1 in. thick.

12 board ft. = 1 cu. ft.

Board feet $\div$ 12 = cu. ft.

Cu. ft. $\times$ 12 = board ft.

To find surface in square feet, all dimensions being in feet, multiply length by breadth. When either dimension is in inches, multiply as above, and divide the product by 12. When all dimensions are in inches, multiply as above, and divide by 144.

To find volume of square cut timber, all dimensions being in feet, multiply length, breadth and depth; the result will be the volume in cubic feet. When one dimension is in inches, multiply as above, and divide by 12. When two dimensions are in inches, multiply as above, and divide by 144. When all three dimensions are in inches, multiply as above, and divide by 1,728.

To find the volume of round timber: When dimensions are in feet, multiply the mean girth by the diameter, divide by 4, and multiply by the length. The result gives the cubic feet. When the length is in feet, girth and diameter in inches, proceed as above, and divide the product by 144. When all dimensions are in inches, proceed as in the first case, and divide the product by 1,728.

Stone and Masonry Measure.

Masonry and stone are measured by the cubic foot and by the perch, or, superficially, by the square foot or square yard. A

perch is 16½ feet long, 1 1-2 feet wide, 1 foot high, and contains 24.75 cubic feet. In measuring a stone wall a perch is often figured at 22 cubic feet of stone, 2.75 being allowed for mortar, and in some of the New England states a perch is called 16½ cubic feet. Excavations are measured by the cubic yard; 1 cubic yard is called a load. Brickwork is figured by the thousand of bricks, or in cubic feet.

METRIC SYSTEM.

The metric system is compulsory in France, Germany, Austria-Hungary, Belgium, Spain, Portugal, Italy, Norway, Sweden, Switzerland, Servia, Roumania, Mexico, Brazil, Peru, Venezuela and the Argentine Republic. It is legalized but not compulsory in the United States (act of 1866), Great Britain, Denmark and Japan. In the United States, however, and in Great Britain, both custom and the greater convenience of computation—aside from notation—in the duodecimal over the decimal system have prevailed to maintain the old customary standards in common use. The Federal government has furnished exact metric standards to the several states. The metric system is used extensively in scientific work, as it requires no adaptation of different national standards, and thus facilitates the mutual exchange of scientific research in different countries.

By convention among the leading nations of the world an international bureau of weights and measures has been created with its seat near Paris, which has prepared two ingots of pure platinum-iridium. From one of these a number of standard kilograms has been made, and from the other a number of standard meter bars, both derived from the standards in the French archives. Certain of these copies were preserved as intenational standards, and others were distributed among the contributory governments. Those sent to the United States are in the possession of the Coast and Geodetic Survey. These copies of the international prototype meter and kilogram form the basis of official metrology in the United States.

The metric system employs the decimal notation, being based upon decimal calculation entirely. The primary unit is the meter for the measure of length. It was intended to be the ten-millionth

part($\frac{1}{10,000,000}$) of the earth's quadrant, i. e., of that part of a meridian from either pole to the equator. After this length was obtained and a set of standards prepared and deposited in the archives of France, it was discovered that errors were made in the calculations. Nevertheless, the standards were left undisturbed. Derivative units are: *Are* fot surface, *liter* for capacity, *gram* for weight. Denominations of decimal fractional parts, or decimal multiples of the units are formed by prefixing Greek numerals for the higher denominations, as dekameter = 10 meters, hektometer = 100 meters, kilometer = 1,000 meters, myriameter = 10,000 meters; or Latin numerals for the lower denominations, as decimeter = 0.1 meter, centimeter = 0.01 meter, millimeter = 0.001 meter.

By the Act of Congress of July 28, 1866, the relation 1 meter = 39.37 inches was legalized. In Great Britain, the same meter is equal to 39.370113 inches. The difference in value between the British and U. S. inch, foot and yard, is thus so slight that for practically all purposes, the difference is negligible.

The legal weight of a gram in the United States is 15.432 grains. It is the weight of a cubic centimeter of pure water at 39.2° F.

In addition to the standards mentioned, there is current a variety of customary measures of different denominations and irregular values, which are given under the several heads in the succeeding pages.

Elements of the Metric System.

Length.	Surface.	Capacity.	Weight.	Notation.
			Metric Ton	1000000.
			Quintal	100000.
Myriameter			Myriagram	10000.
Kilometer		Kiloliter	Kilogram	1000.
Hektometer	Hektare	Hektoliter	Hectogram	100.
Dekameter	Dekare	Dekaliter	Dekagram	10.
Meter.	Are.	Liter.	Gram.	1.
Decimeter		Deciliter	Decigram	0.1
Centimeter	Centiare	Centiliter	Centigram	0.01
Millimeter		Milliliter	Milligram	0.001
Micromillimeter				0.000001

Measures of Length.

1 Millimeter (mm.) = 1,000 micromillimeters.
1 Centimeter (cm.) = 10 millimeters.
1 Decimeter (dm.) = 10 centimeters.
1 Meter (m) = 10 decimeters.
1 Dekameter (Dm) = 10 meters.
1 Hektometer (hm) = 10 dekameters.
1 Kilometer (km.) = 10 hektometers.
1 Myriameter (Mm.) =10 kilometers.

1356 mm. = 135.6 cm. = 13.56 dm. = 1.356 m.= 0.1356 Dm.

The micromillimeter is used in microscopical measures.

The millimeter is used in mechanical and scientific measures.

The meter is used in measuring short distances.

The kilometer is used in measuring long distances (roads, etc.)

Measures of Surface.

The units of square measure are squares, the sides of which are equal to a unit of long measure.

1 sq. mm.
1 sq. cm. = 100 sq. millimeters.
1 sq. dm. = 100 sq. centimeters.
1 sq. m., or 1 centiare (ca.) = 100 sq. decimeters.
1 sq. Dm., or are = 100 square meters.
1 sq. hm., or 1 hektare (ha.) = 100 sq. dekameters.
1 sq. km. = 100 sq. hektometers.

Measures of Volume.

The units are cubes, the edges of which are equal to a unit of long measure.

1 cu. centimeter (c. cm. or c.c.) = 1,000 cu. millimeters (c. mm.).
1 cu. decimeter (c. dm.) = 1,000 cu. centimeters.
1 cu. meter (c. m.) = 1,000 cu. decimeters.
1 cubic decimeter, liquid or dry measure = 1 liter (l.).

Wood Measure.

1 cubic meter or stere = 10 decisteres = 1,000 cu. decimeters = 0.2759 cord, or 35.3165 cu. feet.

1 dekastere = 10 steres = 2.759 cords.

Measures of Capacity.

The unit of capacity is the liter, for liquid and dry measure, being equal to a cubic decimeter of water, which at maximum density will counterpoise the standard kilogram in vacuum.

The hektoliter is the unit in measuring large quantities of grain, fruits, liquids (beer), etc.

1 Centiliter (cl.) = 10 milliliters.
1 Deciliter (dl.) = 10 centiliters.
1 Liter (l.) = 10 deciliters.
1 Dekaliter (Dl.) = 10 liters.
1 Hektoliter (hl.) = 100 liters.
1 Kiloliter (kl.) or stere = 1,000 liters.
1 Myrialiter (Ml.) = 10,000 liters.

Measures of Weight.

1 Milligram (mg).
1 Centigram (cg.) = 10 milligrams.
1 Decigram (dg.) = 10 centigrams.
1 Gram (g.) = 10 decigrams.
1 Dekagram (Dg.) = 10 grams.
1 Hektogram (hg.) = 10 dekagrams.
1 Kilogram (kg.) = 10 hektograms.
1 Myriagram (Mg.) = 10 kilograms.
1 Quintal or meter hundredweight (Q.) = 10 myriagrams or 100 kilograms (kilos).
1 Tonneau or metric ton or millier = 10 quintals or 1,000 kilos.

The kilogram is the common commercial weight in countries using the metric system, and is the weight of a liter (cu. dm.) or water at greatest density.

Converting Metric to Common Measure and Vice Versa.

To change the metric to the common system, reduce the metric number to the denomination of the principal unit of the table; then multiply by the equivalent in the common system and reduce the product to the required denomination.

Example.—How many feet are equal to 4.6 kilometers?

4.6 km. × 1,000 = 4,600 m.
39.37 in. × 4,600 = 181,102 inches = 1,508.5 feet.

To change the common to the metric system, reduce the

given quantity to the denomination in which the equivalent of the principal unit of the metric table is expressed; then divide by this equivalent and reduce the quotient to the required denomination.

Example.—12 lbs. 6 oz. Troy = how many kilograms?

12 lbs. 6 oz. = 12.5 lbs.
12.5 lbs. × 5,760 grains = 72,000 grains.
72,000 grains ÷ 15.432 grains Tr. = 4,665.63 grams.
4,665.63 grams ÷ 1,000 = 4.66563 kg.

Approximate Equivalents of Common and Metric Measures.

Long Measure.

1 kilometer = 0.625 mile.
1 mile = 1.6 kilometers.
1 pole or perch = 5 meters.
1 chain = 20 meters.
1 furlong = 200 meters.
5 furlongs = 1 kilometer.
1 foot = 3 decimeters = 30 centimeters.
1 meter = 3.280833 feet = 3 feet 3⅜ inches.
11 meters = 12 yards.
1 decimeter = 4 inches.
1 centimeter = 2/5 inch.
1 millimeter = 1/25 inch.

To convert meters into inches, multiply by 39.37.

To convert inches into meters, divide by 39.37.

Square Measure.

1 square inch = 6.5 square centimeters.
1 square meter = 10.75 square feet.
1 acre = 1.16 per cent over 4,000 square meters.
1 square mile = 259 hektares.

Measures of Volume.

1 gallon = 3.78 liters.

1 liter = 0.265 gallon = 1.06 liquid quarts = 2.11 pints.

1 cubic foot = 28.3 liters.

1 cubic meter = 1.31 cubic yards.

1 cubic yard = 0.75 cubic meter.

1 kiloliter of water = 2,204 pounds.

1 hektoliter = 2.8 bushels = 26.4 gallons = 0.8522 beer barrel = 0.8387 standard barrel.

Measures of Weight.

1 long ton = 1,016 kilograms.
1 gram = 15.43 grains.
1 kilogram = 2.2 pounds avoirdupois.
1,000 kilograms (1 metric ton) = 0.984 long ton.

Miscellaneous.

1 Gunter's chain	= 20.1168 meters.
1 sq. statute mile	= 259.000 hectares.
1 fathom	= 1.829 meters.
1 nautical mile	= 1853.25 meters.
1 foot	= 0.304801 meter.
1 avoirdupois pound	= 453.5924277 grams.
15,432.35639 grains	= 1 kilogram.
1 inch	= 25.4 millimeters.

COMPARATIVE TABLES OF COMMON AND METRIC MEASURES.

Long Measure.

Measures.	Feet.	Inches.	Miles.	Meters.	Kilometers
Centimeter	0.03280	0.3937	....	0.01	.....
Chain	.....	.....	.....	20.117	0.0201
Decimeter	0.32808	3.937	.....	0.1	.....
Dekameter	32.80833	393.7	.....	10.	.....
Fathom	6.	72.	.....	1.8288	.....
Foot	1.	12.	.....	0.3048	.....
Furlong	660.	7920.	0.125	201.16	0.20116
Hand	0.3333	4.	.....	0.1016	.....
Hektometer	328.0833	.....	0.0621	100.	0.1
Inch	0.0833	1	.....	0.0254	.....
Kilometer	32-0.833	.....	0 6213	1000.	1.
League	18241.	.....	3.45	4827.9	4.8279
Line	.....	0.0833	.....	0.002116	.....
Link	0.666	8.	.....	0.2032	.....
Meter	3.28083	39.37	0.00062	1.	0.001
Mile, Statute	5280.	63360.	1.	1609.	1.6093
Millimeter	0.00328	0.03937	.....	0.001	.....
Myriameter	.....	.....	6.21:8	10000.	10.
Perch, Pole, Rod	16.5	198.	.....	5.0290	0.005
Span	0.75	9.	.....	0.2286	.....
Yard	3.	36.	0.0006	0.9144	0.0009

Square Measures.

Measures.	Sq. In.	Sq. Feet.	Sq. Yds.	Acres.	Sq. Meters.
Sq. Centimeter	0.155	0.00107641	.....	.....	0.0001
Sq. Dekam't'r. Are	.....	1.0764101	119.601	0.0247	100.
Sq. Decimeter	15.5	0.10764101	.....	.....	0.01
Sq. Foot	144.	1.	0.111	.....	0 0929
Sq. Kilometer	.....	0.3861 sq. mi.	.....	247.	1000000.
Sq. M't'r (Centiare)	1550	10.764101	1.19601	.....	1.
Sq. Mile	.....	.....	3097600	640.	.....
Sq. Millimeter	0.00155	0.00001076	.....	.....	0.000001
Sq. Rod	.....	272.225	30.25	0.00625	25.293
Sq. Yard	1296.	9.	1.	0.0002	0.8361
Acre	.....	43560.	4840.	1.	4046.86
Dekare	.....	10764 1	1196.01	0.2471	1000.
Hektare	.....	107641.	11960.11	2.4711	10000.
Square Inch	1.	0 007	.....	.....	0.00065

Cubic Measures (Measures of Volume).

Measures.	Cubic Feet, Inches, or Yards.		Metric.	
Cubic centimeter	0.061025	cu. in.	1.	c. cm.
Cubic decimeter	61.0254	cu. in. or		
	0.0353	cu. ft.	1.	c. dm.
Cubic foot	1728.	cu. in.	28.3161	c. dm.
Cubic inch	1.	cu. in.	16.3866	c. cm.
Cubic meter, or Stere	35.3156	cu. ft. or		
	1.3080	cu. yd.	1.	c. m.
Cubic millimeter	0.00610	cu. in.	1.	c. mm.
Cubic yard	27.	cu. ft.	0 76453	c. m.

Equivalents of Liquid Measures in Common and Metric Measures and Cubic Feet or Inches.

Measures.	U. S. Gallons.	Cubic Feet.	British Imperial Gallons.	Liters.
Barrel, Beer, U. S.	31.	4.144	25.8138	117.3483
Barrel, Standard, U. S.	31.5	4.211	26.2301	119.2410
Barrel, Beer, British	43.2328	5.7795	36.	163.6547
Centiliter	0.0026417	61.0234*	0.00219975	0.01
Dekaliter	2.6417	0.353145	2.19975	10.
Deciliter	0.026417	6.10234*	0.0219975	0.1
Fluid dram, U. S.	0.00097656	0.22559*	0.0008132	0.003697
Fluid ounce, U. S.	0.0078125	1.80469*	0.006505	0.02957
Gallon, British	1.20091	277.410*	1.	4.545963
Gallon, U. S.	1.	231*	0.832702	3.78543
Gill, British	0.03753	8.66905*	0.03125	0.142061
Gill, U. S.	0.03125	7.21875*	0.02602	0.118295
Hectoliter	26.417	3.53145	21.9975	100.
Hogshead, British	75.6573	10.1139	63.	286.396
Hogshead, U. S.	63.	8.4219	52.4602	238.48
Kiloliter	264.17	35.3145	219.975	1000.
Liter	0.26417	61.0234*	0.219975	1.
Milliliter	0.00026417	0.0610234*	0.000219975	0.001
Pint, British	0.1501	34.6762*	0.125	0.568245
Pint, U. S.	0.125	28.875*	0.104088	0.473179
Pipe, British	151.314	20.2278	126.	572.791
Pipe, U. S.	126.	16.8438	104.924	476.96
Puncheon, British	102.8764	13.4852	84.	381.861
Puncheon, U. S.	84.	11.2292	69.947	317.976
Quart, British	0.3002	69.3525	0.25	1.13649
Quart, U. S.	0.25	57.750	0.2082	0.946359
Tierce, British	50.4382	6.7426	42.	190.93
Tierce, U. S.	42.	5.6146	34.9735	158.99
Tun, British	302.629	40.4556	252.	1145.58
Tun, U. S.	252.	33.6876	209.841	953.93

* Cubic inches.

Equivalents of Dry Measures in Common and Metric Measures and Cubic Feet or Inches.

Measures.	U. S. Measure.		Cubic Feet or Cubic Inches.		British Measure.		Liters.
Bushel, Brit., Imp.	1.03202	bu.	1.28431	cu. ft.	1.	bu.	36.3677
Bushel, U. S.,Win.	1.	bu.	1.24446	cu. ft.	0.968972	bu.	35.239
Centiliter	0.0181616	pt.	0.610234	cu.in.	0.017598	pt.	0.01
Coomb, British..	4.12808	bu	5.13724	cu. ft.	4.	bu.	145.471
Deciliter	0.181616	pt.	6.10234	cu. in.	0.17598	pt.	0.1
Dekaliter	9.08078	qt.	610.234	cu. in.	8.79902	qt.	10.
Gallon, British ..	0.129002	bu.	277.410	cu. in.	0.125	bu.	4.545963
Half peck	0.125	bu.	268.8025	cu. in.	0.12112	bu.	4.40491
Hectoliter	2.83774	bu.	3.53145	cu. ft.	2.74969	bu.	100.
Kiloliter	28.3774	bu.	35.3145	cu. ft.	27.4969	bu.	1000.
Liter	0.908078	qt.	61.0234	cu. in.	0.879902	qt.	1.
Peck, British ...	1.03202	peck	554.820	cu. in.	1.	peck	9.091926
Peck, U. S.....	1.	peck	537.605	cu. in.	0.968972	peck	8.80982
Pint, U. S.....	1.	pt.	33.6003	cu. in.	0.968972	pt.	0.550614
Quart, U. S.....	1.	qt.	67.2006	cu. in.	0.968972	qt.	1.10123
Quarter, British.	8.25616	bu.	10.27448	cu. ft.	8.	bu.	290.9416
Wey, British	41.2808	bu.	51.3724	cu. ft.	40.	bu.	1454.708

Equivalents of One Cubic Foot in Common and Metric Measures.

Measures.	1 Cubic Foot =
Barrel, U. S., Standard, 31.5 gals.....	0.23748
Barrel, U. S., Beer, 31 gals.....	0.24131
Bushel, struck, U. S., 2150.42 cu. in.....	0.803564
Bushel, British	0.778630
Gallons, U. S., liquid, 231 cu. in.....	7.48052
Gallons, British	6.22905
Half-pecks, U. S.....	6.42851
Inches, cubic	1728.
Liters	28.3170
Pecks, U. S.....	3.21426
Pecks, British	3.11452
Pints, U. S., liquid.....	59.84416
Pints, U. S., dry.....	51.42809
Pints, British	49.8324
Quarts, U. S., liquid.....	29.92208
Quarts, U. S., dry.....	25.71405
Quarts, British	24.9162
Yard, cubic	0.037037

Equivalents of Avoirdupois and Metric Weights.

Measures.	Avoirdupois.	Grams or Kgs.
Centigram	0.15432 grains	0.01 g.
Dekagram	154.32356 grains	10. g.
Decigram	1.54324 grains	0.1 g.
Dram, U. S. and Brit. Avoir.	27.34375 grains	1.7718 g.
Dram, U. S. and Brit. Apoth.	60. grains	3.8879 g.
Grain, U. S. and Brit.	1. grains	0.0648 g.
Gram	15.43236 grains 0.03527 oz.	1. g.
Hektogram	3.5274 oz.	100. g.
Hundredweight, long	112. lbs.	50.8024 kg.
Hundredweight, short	100. lbs.	45.3592 kg.
Kilogram	2.20462 lbs.	1000. g.
Millier, or Tonne	2204.62 lbs.	1000. kg.
Milligram	0.01543 grains	0.001 g.
Myriagram	22.0462 lbs.	10. kg.
Ounce, Avoir., U. S. and Brit.	437.5 grains	28.3495 g.
Ounce, Troy and Apoth., U. S. and Brit.	480.000 grains	31.1035 g.
Pennyweight, Troy, U. S. and Brit.	24. grains	1.5552 g.
Pound, U. S. and Brit. Avoir.	1. lb. 7000. grains	453.5924 g. 0.4536 kg.
Pound, U. S. and Brit. Troy and Apoth.	0.82286 lbs. 5760. grains	373.242 g. 0.3732 kg.
Quarter, long	28. lbs.	12.7006 kg.
Quarter, short	25. lbs.	11.3398 kg.
Quintal, U. S. and Brit.	100. lbs.	45.3592 kg.
Quintal, French	220.462 lbs.	100. kg.
Scruple, U. S. and Brit.	20. grains	1.2960 g.
Stone, Brit.	14. lbs.	6.3503 kg.
Ton, U. S. and Brit., long	2240. lbs.	1016.005 kg.
Ton, U. S., short	2000. lbs.	907.185 kg.
Tonneau, French	2204.62 lbs.	1000. kg.

A Cubic Inch is Equal to

16.38716 milliliters; or 1.638716 centiliters; or 0.1638716 deciliter; or 0.01638716 liter; or 0.0005787 cubic ft.; or 0.138528 U. S. gill; or 1.90985 spherical inches.

A Cubic Yard is Equal to

27 cu. ft., or 201.974 U. S. gals.
46,656 cu. in.
0.0764559 myrialiter.
0.764559 kiloliter, or cu. meter.
7.64559 hectoliters.
7.2 flour barrels of 3 struck bus.
76.4559 decaliters.
764.559 liters, or cu. decimeters.
7,645.59 deciliters.
21.69623 U. S. bushels (struck).
21.0230 British bushels.

Compound Units.

1 lb. per sq. inch (453.59 g.) =
- 144 lbs. per square foot.
- 2.0355 in. of mercury at 32° F.
- 2.0416 in. of mercury at 62° F.
- 2.309 ft. of water at 62° F.
- 27.71 in. of water at 62° F.
- 0.07031 kg. per sq. cm.

1 atmosphere (14.7 lbs., or 6.6679 kg. per sq. in.) =
- 2116.3 lbs. per square foot.
- 33.947 ft. of water at 62° F.
- 30 in. of mercury at 62° F.
- 29.922 in. of mercury at 32° F.
- 760 millimeters of mercury at 32° F.

1 inch of water at 62° F. =
- 0.0361 lb. per square inch.
- 5.196 lbs. per square foot.
- 0.0736 in. of mercury at 62° F.

1 inch of water at 32° F. =
- 5.2021 lbs. per square foot.
- 0.036125 lb. per square inch.

1 foot of water at 62° F. =
- 0.433 lb. per square inch.
- 62.355 lbs. per square foot.
- 0.883 in. of mercury at 62° F.

1 inch of mercury at 62° F. (2.54 cm.) =
- 0.49 lb. per square inch.
- 70.56 lbs. per square foot.
- 1.132 ft. of water at 62° F.
- 13.58 in. of water at 62° F.

1 kilogram per square inch = 317.46 lbs. per square foot.

1 kilogram per square centimeter = 14.223 lbs. per sq. inch.

Measures of Time.

60 seconds = 1 minute.
60 minutes = 1 hour.
24 hours = 1 day.
365 days = 1 common year.
366 days = 1 leap year.

In civil computation, i. e., for all the practical purposes of life, the day commences at midnight and is divided into two portions of 12 hours each, from midnight to noon, and from noon to midnight.

In nautical time and astronomical computation the day begins at noon. In nautical time the day is divided into watches of 4 hours each. In astronomical time the day is counted through the 24 hours.

A *solar day* is measured by the rotation of the earth upon its axis with respect to the sun.

A *solar year* is the time in which the earth makes one revolution around the sun. Its average time, or the *mean solar year,* is 365 days, 5 hours, 48 minutes, 49.7 seconds, or nearly 365¼ days. A *mean lunar month* is 29 days, 12 hours, 44 minutes and 3.84 seconds.

By the *Julian* (or old style) *Calendar,* introduced by Julius Cæsar, the year was taken as 365 days, 6 hours. To equalize the loss of time, one extra day was inserted every fourth year (leap year), being placed at the end of February. The slight difference between the length of the year assumed in the Julian Calendar and the actual length of the year, amounting to 11 minutes, 12 seconds every year, had accumulated to a period of 10 days at the time of Pope Gregory XIII, who suppressed 10 days out of the year 1582, going from October 5 to 15. In the Reformed or Gregorian (new style) Calendar, the repetition of this accumulation of error is prevented by leaving out 3 of the extra days every 400 years, making this omission in the years which are not exactly divisible by 400. Thus of the leap years 1700, 1800, 1900, 2000, only the last is made leap year. This Gregorian Calendar was introduced in England in 1752, when 11 days were omitted. It is now in force in all Christian countries, except Russia, which is 12 days behind the rest in its time.

Legal Units of Electrical Measure.

In accordance with a resolution adopted by the International Electrical Congress, held at Chicago, in 1893, the Congress of the United States has established the following units of electrical measure:

First. The unit of resistance shall be what is known as the international *ohm,* which is substantially equal to one thousand million units of resistance of the centimeter-gram-second system of electro-magnetic units, and is represented by the resistance offered to an unvarying electric current by a column of mercury at the temperature of melting ice fourteen and four thousand five hundred and twenty-one ten-thousandths grams in mass, of a constant cross-sectional area, and of the length of one hundred and six and three-tenths centimeters.

Second. The unit of current shall be what is known as the international *ampere,* which is one-tenth of the unit of current

of the centimeter gram-second system of electro-magnetic units, and is the practical equivalent of the unvarying current, which, when passed through a solution of nitrate of silver in water in accordance with standard specifications, deposits silver at the rate of one thousand one hundred and eighteen millionths of a gram per second.

Third. The unit of electromotive force shall be what is known as the international *volt,* which is the electromotive force that, steadily applied to a conductor whose resistance is one international ohm, will produce a current of an international ampere, and is practically equivalent to one thousand fourteen hundred and thirty-fourths of the electromotive force between the poles or electrodes of the voltaic cell known as Clark's cell, at a temperature of fifteen degrees Centigrade, and prepared in the manner described in the standard specifications.

Fourth. The unit of quantity shall be what is known as the international *coulomb,* which is the quantity of electricity transferred by a current of one international ampere in one second.

Fifth. The unit of capacity shall be what is known as the international *farad,* which is the capacity of a condenser charged to a potential of one international volt by one international coulomb of electricity.

Sixth. The unit of work shall be the *joule,* which is equal to ten million units of work in the centimeter-gram-second system, and which is practically equivalent to the energy expended in one second by an international ampere in an international ohm.

Seventh. The unit of power shall be the *watt,* which is equal to ten million units of power in the centimeter-gram-second system, and which is practically equivalent to the work done at the rate of one joule per second.

Eighth. The unit of induction shall be the *henry,* which is the induction in a circuit when the electromotive force induced in this circuit is one international volt, while the inducing current varies at the rate of one ampere per second.

Mathematics and Tables

Arithmetical and Algebraical Signs and Abbreviations.

There are a number of signs used in engineering calculations, and the following gives a brief description of them.

The meaning of the signs + (plus), — (minus), × (times), and ÷ (divided by), is indicated by the following examples:

$$3 + 5 = 8$$
$$5 - 2 = 3$$
$$7 \times 4 = 28$$
$$54 \div 9 = 6$$

Parentheses ( ), or *brackets* [ ] denote that all that is included in them is to be considered as one quantity. When the times (×) sign is used in a calculation, it unites the figures on either side, the same as parentheses, and the operation thus indicated must be performed first. Thus in the formulas

$$(a + b) \times c \text{ and } a + b \times c$$

if a, b and c are equal to 7, 3, and 8 respectively, then

$$(7 + 3) \times 8 = 10 \times 8 = 80$$

and

$$7 + 3 \times 8 = 7 + 24 = 31$$

Involution.—7^2 reads seven squared, and means 7×7.

7^3 reads seven cubed, and means $7 \times 7 \times 7$.

7^4, 7^5, etc., read seven raised to the fourth, fifth, etc., power, and mean $7 \times 7 \times 7 \times 7$, and $7 \times 7 \times 7 \times 7 \times 7$, etc.

Evolution.—$\sqrt{9}$ reads the square root of nine, and means the figure which squared gives nine. In this case that figure is 3.

$$\sqrt{9} = 3 \text{ because } 3^2 = 3 \times 3 = 9.$$

$\sqrt[3]{125}$ reads the cube root of one hundred and twenty-five, and means the figure which cubed gives 125. In this case that figure is 5.

$\sqrt[3]{125} = 5$, because $5^3 = 5 \times 5 \times 5 = 125$.

$\sqrt[4]{N}$, $\sqrt[5]{N}$, etc., read the 4th root, the 5th root, etc., of a certain number, and mean the figure which raised to the 4th, 5th, etc., power gives this number.

$$\sqrt[7]{128} = 2 \text{ because } 2^7 = 128.$$

Instead of using the designations $\sqrt{N}$, $\sqrt[3]{N}$, $\sqrt[4]{N}$, etc., roots are sometimes expressed in the following way:

$N^{\frac{1}{2}}$, $N^{\frac{1}{3}}$, $N^{\frac{1}{4}}$, etc., which read N raised to the power one-half, one-third, etc., but mean the same as square root, cube root, etc. Similarly $N^{\frac{5}{3}}$ means, that the figure should be raised to the fifth power, and thereafter the cube root taken.

$$8^{\frac{2}{3}} = \sqrt[3]{8^2} = \sqrt[3]{64} = 4.$$

$$12^{1.3} = 12^{\frac{13}{10}} = \sqrt[10]{12^{13}}.$$

Expressions such as 7^{-3}, 10^{-9}, $N^{-\frac{1}{2}}$, are sometimes contained in formulas, and mean the same as $\frac{1}{7^3}$, $\frac{1}{10^9}$, $\frac{1}{N^{\frac{1}{2}}}$

$$96 \times 4^{-3} = 96 \times \frac{1}{4^3} = \frac{96}{4^3} = 96 \div 4^3 = 96 \div 64 = 1\tfrac{1}{2}.$$

$$9^{-\frac{3}{2}} = \frac{1}{9^{\frac{3}{2}}} = \frac{1}{\sqrt{9^3}} = \frac{1}{\sqrt{729}} = \frac{1}{27}$$

Cancellation.

Cancellation is very helpful in many calculations. The principle of it may easily be understood from the following example:

$$\frac{32 \times 6 \times 7 \times 11}{12 \times 44 \times 14 \times 9}$$

Any figure of the numerator and any figure of the denominator may be divided by the same figure. In this instance, the expression is cancelled in the following way:

7 in the numerator, and 14 in the denominator are divided by 7. Again 6 and 2 are divided by 2; 3 and 12 are divided by 3; 32 and 44 are divided by 4. We now have 11 both in numerator and

denominator and may cancel both. 8 and 4 are divided by 4. The expression is simplified to:

$$\frac{2 \times 1 \times 1 \times 1}{1 \times 1 \times 1 \times 9} = \frac{2}{9}$$

When numerator or denominator is a sum or difference of quantities, cancellation cannot be used.

Transposing Equations.

The principle in transposing an equation is that the same operation may be performed on each side of the equation, i. e., we may add the same quantity to each side, subtract the same quantity from each side, multiply each side by the same quantity, divide each side by the same quantity.

For instance, in the equation $A = B + C$, B may be transposed to the other side of the equation by subtracting B from each side of the equation, thus: $A - B = B + C - B$, and since $B - B = O$, then $A - B = C$.

If we have $A \times B = C$, and wish to transpose B to the other side, we divide each side of the equation by B, thus:

$$\frac{A \times B}{B} = \frac{C}{B} \text{ ; hence } A = \frac{C}{B}$$

If we have $\frac{A}{B} = C$, and desire to transpose B to the other side of the equation, we multiply each side by B, thus:

$\frac{A \times B}{B} = B \times C$; cancelling the B's, we obtain: $A = B \times C$.

These simple illustrations form the foundation for more complex transpositions such as are frequently used in calculations, and several instances of which are found in this book.

Interpolation of Tables.

A table cannot be expected to contain the data corresponding to every figure we happen to look for. If the figure desired is not found directly in the table, and a very accurate figure is desired, it may be found by a method of calculation called interpolation. In

most cases an estimate sufficiently accurate for all practical purposes may be made.

A few examples of interpolation will explain the method best.

Example 1.—What is the latent heat of ammonia at 22 pounds gauge pressure?

Solution.—Referring to the ammonia table, page 490, we take the two nearest pressures above and below the desired pressure, in this case 21.63 and 22.48 pounds.

Latent heat at	21.63 pounds =	563.878
Latent heat at	22.48 pounds =	563.113
Difference in latent heat for	0.85 pound =	0.765

Since $22 - 21.63 = 0.37$, we find the difference in latent heat for this difference in pressure by proportion:

$$0.85 : 0.765 =$$
$$0.37 : ?$$
$$\frac{0.37 \times 0.765}{0.85} = 0.333$$

The latent heat at 22 pounds gauge pressure is therefore $563.878 - 0.333 = 563.545$.

Example 2.—How many pounds of sodium chloride are required per cubic foot of solution, in order to prepare an 18% solution?

Solution.—From the table, page 591, we find the pounds required for 15% and 20% to be 10.40 and 14.40 respectively. Thus for a difference of $20 - 15\%$, there is a difference in pounds required of $14.40 - 10.40 = 4$ pounds. For $18 - 15$ or 3% the difference can then be found by proportion, thus:

$$5 : 4 =$$
$$3 : ?$$
$$\frac{3 \times 4}{5} = 2.4$$

The pounds required for an 18% solution are therefore $10.40 + 2.4 = 12.80$ pounds.

Logarithms.

It has been calculated that $10^{0.30103} = 2$; $10^{0.47712} = 3$; $10^{0.60206} = 4$; $10^{2.0} = 100$; $10^{2.00043} = 101$; $10^{2.00087} = 102$; etc., and the word *logarithm* has been introduced, the meaning of

which will be understood from the following expressions taken in connection with the above equations:

0.30103 is the logarithm of 2, written log 2.
0.47712 is the logarithm of 3, written log 3.
2.0 is the logarithm of 100, written log 100.
2.00043 is the logarithm of 101, written log 101.

Instead of using 10 as the base for logarithms there are mathematical reasons for using, in certain cases, the figure 2.71828. On this base, the result of the calculation is as follows: $2.71828^{0.6931} = 2$; $2.71828^{1.0986} = 3$; $2.71828^{1.3863} = 4$; etc. The logarithms calculated on this base are called *hyperbolic logarithms;* thus:

0.6931 is the hyperbolic logarithm of 2, written hyp. log. 2.
1.0986 is the hyperbolic logarithm of 3, written hyp. log. 3.
1.3863 is the hyperbolic logarithm of 4, written hyp. log. 4.

Wherever in this book the word logarithm, or the abbreviation log., is used without specifying a certain system, the common logarithm with 10 as base is always meant.

Logarithms enter directly into some important formulas for work and conditions in engine and compressor cylinders. They are also used for finding the roots of a number, and for raising a figure to a high power, and for performing long multiplications and divisions.

On pages 815 to 834 are given the ordinary logarithms (with 10 as base), and on page 835 are given the hyperbolic logarithms.

The logarithm of 1 is 0 in any system; the logarithm of the base is 1. In a system where the base is greater than 1, the logarithms of all numbers above 1 are positive, those below 1 are negative.

The unit or integral part of a logarithm is called the *index* or *characteristic,* the decimal part the *mantissa.*

In the usual tables of common logarithms only the mantissa is given, with the decimal point in front omitted. To find the index take the number of digits to the left of the decimal point less 1. From 0 to 10 it is 0, from 10 to 99 it is 1, from 100 to 999 it is 2, etc. For instance, the logarithm of 1,350 is 3.130334. In the table, the mantissa 130,334 is found, and that part of the logarithm is the same for 1,350, 13,500, 135,000, etc., and for 135, 13.5, 1.35, 0.135, 0.0135, etc., the logarithms of these figures

being, in the order mentioned: 3.130334, 4.130334, 5.130334, 2.130334, 1.130334, 0.130334, 0.130334 — 1, 0.130334 — 2, etc. It is easily seen how the index is found by the above rule, as long as the figure for which we want the logarithm is higher than 1.

The index of a logarithm of a decimal fraction is a negative number and is equal to the number of places which the first figure of the decimal other than 0 is removed from the decimal point. Index of log 0.005 = —3 or $\bar{3}$, the minus sign often being placed above the figure. The mantissa is always positive, even though the index be negative.

The *difference* is the tabular difference between the two nearest logarithms.

The *proportional part* is the difference between the given and the nearest smaller tabular logarithm.

Rules for Using Table of Logarithms.

To find the logarithm of a whole number:

For 1 to 9 inclusive. Find the mantissa, as shown below, for 200, 300, etc., up to 900, and prefix the index according to the above rule.

For 10 to 99 inclusive. Again from the rule it is known that for all these figures the index is 1. In order to find the mantissa for 13, find it either for 130 or 1,300 in the way given below.

For 100 to 999. Find the given number in first column of table. Take decimal part of logarithm from column under 0, including the two figures to the left of it, making six figures; prefix the index 2.

Example.—What is the logarithm of 742?

Solution.—First put down the index 2 and the decimal point. Next look up 742, and the mantissa 870404 is found in the 0 column. Thus, log. 742 = 2.870404.

For 1,000 to 9,999. Find the three left-hand or first figures of the given number in No. column and follow the horizontal row to the column indicated by the fourth figure of the given number (including the two figures to the left of the 0 column); prefix the index 3.

Example.—What is the logarithm of 7,423?

Solution.—First, put down the index 3 and the decimal point. As in the example above we again find 742 in the table, then follow

along the horizontal row to the 3 column and obtain here the rest of the mantissa, 870579. Thus:

$$\log. 7{,}423 = 3.870579.$$

For numbers consisting of five digits or more. Find the logarithm for a number composed of the first four figures, as above; take tabular difference from last column and multiply it by the fifth; fifth and sixth; fifth, sixth and seventh, etc., whatever the excess may be over four figures; divide the product by 10, if the given number has 5 figures, by 100 if the given number has 6 figures, and use as the result the nearest whole number. If this has one figure, write it over the last figure of the logarithm of the substituted number, and add; if it has two figures, write it over the last two figures, and add; and if it has three figures (the highest it can have) write it over the three last figures of the said logarithm, and add. This process is called interpolation.

Example.—What is the logarithm of 74,232?

Solution.—The index is 4. The mantissa of 7,423 is 870579. The tabular difference in the D column is 58, which multiplied by the 5th figure, 2, gives 116, and $\frac{116}{10} = 11.6$, the nearest whole number being 12. Adding this 12 to the last two figures of the logarithm, we obtain: $4.870579 + 0.000012 = 4.870591 = \log 74{,}232$.

For a mixed number: Find mantissa as if it were an integer or whole number and prefix the index of the integral part of the number.

For a decimal fraction: Find logarithm as if the figures were all integers, and prefix index according to rule.

Example.—What is the logarithm of 742.32?.

Solution.—It will be noticed that, disregarding the decimal point, this is the same number as in the preceding example, and therefore the mantissa must be the same. The index is 2 (3 digits to the left of the decimal point, index $= 3 - 1$); then:

$$\log. 742.32 = 2.870591.$$

Example.—What is the logarithm of 0.00315?

Solution.—There are 2 ciphers between the decimal point and the first figure higher than 0, then the number of the negative

index is: $2 + 1 = 3$. The mantissa is the same as for the whole number 315. Hence:

$$\text{log. } 0.00315 = 0.498311 - 3.$$

For a vulgar fraction: Reduce to decimal fraction and proceed as for the latter; or subtract logarithm of denominator from that of numerator, the difference being the required logarithm.

Example.—What is the logarithm of $\frac{3}{32}$?

Solution.—Log. $3 = 0.477121$ which may also be written:

$$\text{log. } 3 = 2.477121 - 2$$
$$\text{log. } 32 = 1.505150$$

By subtraction, $\text{log.}\frac{3}{32} = 0.971971 - 2$

To find the number where the logarithm is known: First, where the given logarithm is contained in the table: Find first two decimal figures of logarithm in column of two figures to left of O column and the other figures in the columns to the right. The first three figures of the number will be found in No. column, the fourth at head of column in which the decimals of the logarithm were found. Point off decimals or prefix ciphers according to the index.

Example.—What is the number corresponding to the logarithm 2.752356?

Solution.—Number 5,654 in the table has the given mantissa. As the index is 2, the number asked for must have 3 digits to the left of the decimal point; the required number therefore is 565.4.

Second, where the given logarithm is not contained in the table, we must resort to interpolation. Find the next smaller logarithm in the table and take the number for it, which will give the first four figures of the required number. Subtract the same logarithm from the given one, multiply by 10, 100, etc., according to the number of places wanted, and divide by the difference found in the last column, opposite the logarithm used. Annex quotient to the four figures first obtained and place decimal point, allowing one figure in excess of the index to the left of the decimal point.

Example.—What is the number corresponding to the logarithm 0.879283—1?

Solution.—The next lower mantissa in the table is 879268, corresponding to 7573. The difference between this mantissa and the given mantissa is 15 (correctly speaking it is $\frac{15}{1{,}000{,}000}$ but is always expressed simply this way), and the tabular difference from the D column is 57. If 5 figures are wanted, we take $\frac{15 \times 10}{57} = 3$ approximately.

This 3 is annexed to 7573, giving according to the rule, the number 0.75733.

Computation by Means of Logarithms.

To *multiply* two numbers, add the logarithms of the numbers, and the sum will be the logarithm of the required number.

To *divide* two numbers, subtract logarithm of divisor from that of dividend, the remainder will be the logarithm of the required number.

To raise a number to a given *power,* multiply the logarithm of the number by the exponent of the power; the product will be the logarithm of the required number.

To find any *root* of a given number, divide the logarithm of the number by the index of the root (3 for root, 4 for fourth root, etc.); the quotient will be the logarithm of the root.

To change from common logarithms to *hyperbolic,* multiply the common logarithm by 2.3025851; for practical purposes in most cases 2.3 may be used.

Tables of Common Logarithms (1 to 10,000).

1	.0	26	1.414 973	51	1.707 57	76	1.880 814
2	.301 03	27	1.431 364	52	1.716 003	77	1.886 491
3	.477 121	28	1.447 158	53	1.724 276	78	1.892 095
4	.602 06	29	1.462 398	54	1 732 394	79	1.897 627
5	.698 97	30	1.477 121	55	1 740 363	80	1.903 09
6	.778 151	31	1.491 362	56	1 748 188	81	1.908 485
7	.845 098	32	1.505 15	57	1.755 875	82	1.913 814
8	.903 09	33	1.518 514	58	1.763 428	83	1.919 078
9	.954 243	34	1 531 479	59	1 770 852	84	1.924 279
10	1	35	1 544 068	60	1.778 151	85	1.929 419
11	1.041 393	36	1.556 303	61	1.785 33	86	1.934 498
12	1.079 181	37	1.568 202	62	1 792 392	87	1.939 519
13	1.113 943	38	1.579 784	63	1 799 341	88	1.944 483
14	1.146 128	39	1.591 065	64	1.806 18	89	1.949 39
15	1.176 091	40	1.602 06	65	1.812 913	90	1.954 243
16	1.204 12	41	1.612 784	66	1.819 544	91	1.959 041
17	1.230 449	42	1.623 249	67	1.826 075	92	1.963 788
18	1.255 273	43	1.633 468	68	1.832 509	93	1.968 483
19	1.278 754	44	1.643 453	69	1.838 849	94	1.973 128
20	1.301 03	45	1.653 213	70	1.845 098	95	1.977 724
21	1.322 219	46	1.662 758	71	1.851 258	96	1.982 271
22	1.342 423	47	1.672 098	72	1.857 332	97	1.986 772
23	1.361 728	48	1.681 241	73	1.863 323	98	1.991 226
24	1.380 211	49	1.690 196	74	1.869 232	99	1.995 635
25	1.397 94	50	1.698 97	75	1.875 061	100	2

No.	0	1	2	3	4	5	6	7	8	9	D
100	00- 0000	0434	0868	1301	1734	2166	2598	3029	3461	3891	432
101	00- 4321	4751	5181	5609	6038	6466	6894	7321	7748	8174	428
102	00- 86	9026	9451	9876	—	—	—	—	—	—	425
102	01- —	—	—	—	03	0724	1147	157	1993	2415	424
103	01- 2837	3259	368	41	4521	494	536	5779	6197	6616	420
104	01- 7033	7451	7868	8284	87	9116	9532	9947	—	—	417
104	02- —	—	—	—	—	—	—	—	0361	0775	416
105	02- 1189	1603	2016	2428	2841	3252	3664	4075	4486	4896	412
106	02- 5306	5715	6125	6533	6942	735	7757	8164	8571	8978	408
107	02- 9384	9789	—	—	—	—	—	—	—	—	405
107	03- —	—	0195	06	1004	1408	1812	2216	2619	3021	404
108	03- 3424	3826	4227	4628	5029	543	583	623	6629	7028	400
109	03- 7426	7825	8223	862	9017	9414	9811	—	—	—	398
109	04- —	—	—	—	—	—	—	0207	0602	0998	397
110	04- 1393	1787	2182	2576	2969	3362	3755	4148	454	4932	393
111	04- 5323	5714	6105	6495	6885	7275	7664	8053	8442	883	389
112	04- 9218	9606	9993	—	—	—	—	—	—	—	388
112	05- —	—	—	038	0766	1153	1538	1924	2309	2694	386
113	05- 3078	3463	3846	423	4613	4996	5378	576	6142	6524	383
114	05- 6905	7286	7666	8046	8426	8805	9185	9563	9942	—	383
114	06- —	—	—	—	—	—	—	—	—	032	379

No.	0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.		0	1	2	3	4	5	6	7	8	9	D
115	06-	0698	1075	1452	1829	2206	2582	2958	3333	3709	4083	376
116	06-	4458	4832	5206	558	5953	6326	6699	7071	7443	7815	373
117	06-	8186	8557	8928	9298	9668	—	—	—	—	—	380
117	07-	—	—	—	—	—	0038	0407	0776	1145	1514	370
118	07-	1882	225	2617	2985	3352	3718	4085	4451	4816	5182	366
119	07-	5547	5912	6276	664	7004	7368	7731	8094	8457	8819	363
120	07-	9181	9543	9904	—	—	—	—	—	—	—	362
120	08-	—	—	—	0266	0626	0987	1347	1707	2067	2426	360
121	08-	2785	3144	3503	3861	4219	4576	4934	5291	5647	6004	357
122	08-	636	6716	7071	7426	7781	8136	849	8845	9198	9552	355
123	08-	9905	—	—	—	—	—	—	—	—	—	355
123	09-	—	0258	0611	0963	1315	1667	2018	237	2721	3071	353
124	09-	3422	3772	4122	4471	482	5169	5518	5866	6215	6562	349
125	09-	691	7257	7604	7951	8298	8644	899	9335	9681	—	348
125	10-	—	—	—	—	—	—	—	—	—	0026	346
126	10-	0371	0715	1059	1403	1747	2091	2434	2777	3119	3462	343
127	10-	3804	4146	4487	4828	5169	551	5851	6191	6531	6871	341
128	10-	721	7549	7888	8227	8565	8903	9241	9579	9916	—	338
128	11-	—	—	—	—	—	—	—	—	—	0253	337
129	11-	059	0926	1263	1599	1934	227	2605	294	3275	3609	335
130	11-	3943	4277	4611	4944	5278	5611	5943	6276	6608	694	333
131	11-	7271	7603	7934	8265	8595	8926	9256	9586	9915	—	331
131	12-	—	—	—	—	—	—	—	—	—	0245	330
132	12-	0574	0903	1231	156	1888	2216	2544	2871	3198	3525	328
133	12-	3852	4178	4504	483	5156	5481	5806	6131	6456	6781	325
134	12-	7105	7429	7753	8076	8399	8722	9045	9368	969	—	323
134	13-	—	—	—	—	—	—	—	—	—	0012	323
135	13-	0334	0655	0977	1298	1619	1939	226	258	29	3219	321
136	13-	3539	3858	4177	4496	4814	5133	5451	5769	6086	6403	318
137	13-	6721	7037	7354	7671	7987	8303	8618	8934	9249	9564	316
138	13-	9879	—	—	—	—	—	—	—	—	—	315
138	14-	—	0194	0508	0822	1136	145	1763	2076	2389	2702	314
139	14-	3015	3327	3639	3951	4263	4574	4885	5196	5507	5818	311
140	14-	6128	6438	6748	7058	7367	7676	7985	8294	8603	8911	309
141	14-	9219	9527	9835	—	—	—	—	—	—	—	308
141	15-	—	—	—	0142	0449	0756	1063	137	1676	1982	307
142	15-	2288	2594	29	3205	351	3815	412	4424	4728	5032	305
143	15-	5336	564	5943	6246	6549	6852	7154	7457	7759	8061	303
144	15-	8362	8664	8965	9266	9567	9868	—	—	—	—	302
144	16-	—	—	—	—	—	—	0168	0469	0769	1068	301
145	16-	1368	1667	1967	2266	2564	2863	3161	346	3758	4055	299
146	16-	4353	465	4947	5244	5541	5838	6134	643	6726	7022	297
147	16-	7317	7613	7908	8203	8497	8792	9086	938	9674	9968	295
148	17-	0262	0555	0848	1141	1434	1726	2019	2311	2603	2895	293
149	17-	3186	3478	3769	406	4351	4641	4932	5222	5512	5802	291
150	17-	6091	6381	667	6959	7248	7536	7825	8113	8401	8689	289
151	17-	8977	9264	9552	9839	—	—	—	—	—	—	287
151	18-	—	—	—	—	0126	0413	0699	0986	1272	1558	287
152	18-	1844	2129	2415	27	2985	327	3555	3839	4123	4407	285
153	18-	4691	4975	5259	5542	5825	6108	6391	6674	6956	7239	283
154	18-	7521	7803	8084	8366	8647	8928	9209	949	9771	—	281
154	19-	—	—	—	—	—	—	—	—	—	0051	281
No.		0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.		0	1	2	3	4	5	6	7	8	9	D.
155	19-	0332	0612	0892	1171	1451	173	201	2289	2567	2846	279
156	19-	3125	3403	3681	3959	4237	4514	4792	5069	5346	5623	278
157	19-	59	6176	6453	6729	7005	7281	7556	7832	8107	8382	276
158	19-	8657	8932	9206	9481	9755	—	—	—	—	—	275
158	20-	—	—	—	—	—	0029	0303	0577	085	1124	274
159	20-	1397	167	1943	2216	2488	2761	3033	3305	3577	3848	272
160	20-	412	4391	4663	4934	5204	5475	5746	6016	6286	6556	271
161	20-	6826	7096	7365	7634	7904	8173	8441	871	8979	9247	269
162	20-	9515	9783	—	—	—	—	—	—	—	—	268
162	21-	—	—	0051	0319	0586	0853	1121	1388	1654	1921	267
163	21-	2188	2454	272	2986	3252	3518	3783	4049	4314	4579	266
164	21-	4844	5109	5373	5638	5902	6166	643	6694	6957	7221	264
165	21-	7484	7747	801	8273	8536	8798	906	9323	9585	9846	262
166	22-	0108	037	0631	0892	1153	1414	1675	1936	2196	2456	261
167	22-	2716	2976	3236	3496	3755	4015	4274	4533	4792	5051	259
168	22-	5309	5568	5826	6084	6342	66	6858	7115	7372	763	258
169	22-	7887	8144	84	8657	8913	917	9426	9682	9938	—	257
169	23-	—	—	—	—	—	—	—	—	—	0193	256
170	23-	0449	0704	096	1215	147	1724	1979	2234	2488	2742	255
171	23-	2996	325	3504	3757	4011	4264	4517	477	5023	5276	253
172	23-	5528	5781	6033	6285	6537	6789	7041	7292	7544	7795	252
173	23-	8046	8297	8548	8799	9049	9299	955	98	—	—	251
173	24-	—	—	—	—	—	—	—	—	005	03	250
174	24-	0549	0799	1048	1297	1546	1795	2044	2293	2541	279	249
175	24-	3038	3286	3534	3782	403	4277	4525	4772	5019	5266	248
176	24-	5513	5759	6006	6252	6499	6745	6991	7237	7482	7728	246
177	24-	7973	8219	8464	8709	8954	9198	9443	9687	9932	—	246
177	25-	—	—	—	—	—	—	—	—	—	0176	245
178	25-	042	0664	0908	1151	1395	1638	1881	2125	2368	261	243
179	25-	2853	3096	3338	358	3822	4064	4306	4548	479	5031	242
180	25-	5273	5514	5755	5996	6237	6477	6718	6958	7198	7439	241
181	25-	7679	7918	8158	8398	8637	8877	9116	9355	9594	9833	239
182	26-	0071	031	0548	0787	1025	1263	1501	1739	1976	2214	238
183	26-	2451	2688	2925	3162	3399	3636	3873	4109	4346	4582	237
184	26-	4818	5054	529	5525	5761	5996	6232	6467	6702	6937	235
185	26-	7172	7406	7641	7875	811	8344	8578	8812	9046	9279	234
186	26-	9513	9746	998	—	—	—	—	—	—	—	234
186	27-	—	—	—	0213	0446	0679	0912	1144	1377	1609	233
187	27-	1842	2074	2306	2538	277	3001	3233	3464	3696	3927	232
188	27-	4158	4389	462	485	5081	5311	5542	5772	6002	6232	230
189	27-	6462	6692	6921	7151	738	7609	7838	8067	8296	8525	229
190	27-	8754	8982	9211	9439	9667	9895	—	—	—	—	228
190	28-	—	—	—	—	—	—	0123	0351	0578	0806	228
191	28-	1033	1261	1488	1715	1942	2169	2396	2622	2849	3075	227
192	28-	3301	3527	3753	3979	4205	4431	4656	4882	5107	5332	226
193	28-	5557	5782	6007	6232	6456	6681	6905	713	7354	7578	225
194	28-	7802	8026	8249	8473	8696	892	9143	9366	9589	9812	223
195	29-	0035	0257	048	0702	0925	1147	1369	1591	1813	2034	222
196	29-	2256	2478	2699	292	3141	3363	3584	3804	4025	4246	221
197	29-	4466	4687	4907	5127	5347	5567	5787	6007	6226	6446	220
198	29-	6665	6884	7104	7323	7542	7761	7979	8198	8416	8635	219
199	29-	8853	9071	9289	9507	9725	9943	—	—	—	—	218
199	30-	—	—	—	—	—	—	0161	0378	0595	0813	218
No.		0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.	0	1	2	3	4	5	6	7	8	9	D'
200	30- 103	1247	1464	1681	1898	2114	2331	2547	2764	298	217
201	30- 3196	3412	3628	3844	4059	4275	4491	4706	4921	5136	216
202	30- 5351	5566	5781	5996	6211	6425	6639	6854	7068	7282	215
203	30- 7496	771	7924	8137	8351	8564	8778	8991	9204	9417	213
204	30- 963	9843	—	—	—	—	—	—	—	—	213
204	31- —	—	0056	0268	0481	0693	0906	1118	133	1542	212
205	31- 1754	1966	2177	2389	26	2812	3023	3234	3445	3656	211
206	31- 3867	4078	4289	4499	471	492	513	534	5551	576	210
207	31- 597	618	639	6599	6809	7018	7227	7436	7646	7854	209
208	31- 8063	8272	8481	8689	8898	9106	9314	9522	973	9938	208
209	32- 0146	0354	0562	0769	0977	1184	1391	1598	1805	2012	207
210	32- 2219	2426	2633	2839	3046	3252	3458	3665	3871	4077	206
211	32- 4282	4488	4694	4899	5105	531	5516	5721	5926	6131	205
212	32- 6336	6541	6745	695	7155	7359	7563	7767	7972	8176	204
213	32- 838	8583	8787	8991	9194	9398	9601	9805	—	—	204
213	33- —	—	—	—	—	—	—	—	0008	0211	203
214	33- 0414	0617	0819	1022	1225	1427	163	1832	2034	2236	202
215	33- 2438	264	2842	3044	3246	3447	3649	385	4051	4253	202
216	33- 4454	4655	4856	5057	5257	5458	5658	5859	6059	626	201
217	33- 646	666	686	706	726	7459	7659	7858	8058	8257	200
218	33- 8456	8656	8855	9054	9253	9451	965	9849	—	—	200
218	34- —	—	—	—	—	—	—	—	0047	0246	199
219	34- 0444	0642	0841	1039	1237	1435	1632	183	2028	2225	198
220	34- 2423	262	2817	3014	3212	3409	3606	3802	3999	4196	197
221	34- 4392	4589	4785	4981	5178	5374	557	5766	5962	6157	196
222	34- 6353	6549	6744	6939	7135	733	7525	772	7915	811	195
223	34- 8305	85	8694	888	9083	9278	9472	9666	986	—	194
223	35- —	—	—	—	—	—	—	—	—	0054	194
224	35- 0248	0442	0636	0829	1023	1216	141	1603	1796	1989	193
225	35- 2183	2375	2568	2761	2954	3147	3339	3532	3724	3916	193
226	35- 4108	4301	4493	4685	4876	5068	526	5452	5643	5834	192
227	35- 6026	6217	6408	6599	679	6981	7172	7363	7554	7744	191
228	35- 7935	8125	8316	8506	8696	8886	9076	9266	9456	9646	190
229	35- 9835	—	—	—	—	—	—	—	—	—	189
229	36- —	0025	0215	0404	0593	0783	0972	1161	135	1539	189
230	36- 1728	1917	2105	2294	2482	2671	2859	3048	3236	3424	188
231	36- 3612	38	3988	4176	4363	4551	4739	4926	5113	5301	188
232	36- 5488	5675	5862	6049	6236	6423	661	6796	6983	7169	187
233	36- 7356	7542	7729	7915	8101	8287	8473	8659	8845	903	186
234	36- 9216	9401	9587	9772	9958	—	—	—	—	—	186
235	37- —	—	—	—	—	0143	0328	0513	0698	0883	185
235	37- 1068	1253	1437	1622	1806	1991	2175	236	2544	2728	184
236	37- 2912	3096	328	3464	3647	3831	4015	4198	4382	4565	184
237	37- 4748	4932	5115	5298	5481	5664	5846	6029	6212	6394	183
238	37- 6577	6759	6942	7124	7306	7488	767	7852	8034	8216	182
239	37- 8398	858	8761	8943	9124	9306	9487	9668	9849	—	182
239	38- —	—	—	—	—	—	—	—	—	003	181
240	38- 0211	0392	0573	0754	0934	1115	1296	1476	1656	1837	181
241	38- 2017	2197	2377	2557	2737	2917	3097	3277	3456	3636	180
242	38- 3815	3995	4174	4353	4533	4712	4891	507	5249	5428	179
243	38- 5606	5785	5964	6142	6321	6499	6677	6856	7034	7212	178
244	38- 739	7568	7746	7923	8101	8279	8456	8634	8811	8989	178
No.	0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.		0	1	2	3	4	5	6	7	8	9	D
244	38-	9166	9343	952	9698	9875	—	—	—	—	—	177
245	39-	—	—	—	—	—	0051	0228	0405	0582	0759	177
246	39-	0935	1112	1288	1464	1641	1817	1993	2169	2345	2521	176
247	39-	2697	2873	3048	3224	34	3575	3751	3926	4101	4277	176
248	39-	4452	4627	4802	4977	5152	5326	5501	5676	585	6025	175
249	39-	6199	6374	6548	6722	6896	7071	7245	7419	7592	7766	174
250	39-	794	8114	8287	8461	8634	8808	8981	9154	9328	9501	173
251	39-	9674	9847	—	—	—	—	—	—	—	—	173
251	40-	—	—	002	0192	0365	0538	0711	0883	1056	1228	173
252	40-	1401	1573	1745	1917	2089	2261	2433	2605	2777	2949	172
253	40-	3121	3292	3464	3635	3807	3978	4149	432	4492	4663	171
254	40-	4834	5005	5176	5346	5517	5688	5858	6029	6199	637	171
255	40-	654	671	6881	7051	7221	7391	7561	7731	7901	807	170
256	40-	824	841	8579	8749	8918	9087	9257	9426	9595	9764	169
257	40-	9933	—	—	—	—	—	—	—	—	—	169
257	41-	—	0102	0271	044	0609	0777	0946	1114	1283	1451	169
258	41-	162	1788	1956	2124	2293	2461	2629	2796	2964	3132	168
259	41-	33	3467	3635	3803	397	4137	4305	4472	4639	4806	167
260	41-	4973	514	5307	5474	5641	5808	5974	6141	6308	6474	167
261	41-	6641	6807	6973	7139	7306	7472	7638	7804	797	8135	166
262	41-	8301	8467	8633	8798	8964	9129	9295	946	9625	9791	165
263	41-	9956	—	—	—	—	—	—	—	—	—	165
263	42-	—	0121	0286	0451	0616	0781	0945	111	1275	1439	165
264	42-	1604	1768	1933	2097	2261	2426	259	2754	2918	3082	164
265	42-	3246	341	3574	3737	3901	4065	4228	4392	4555	4718	164
266	42-	4882	5045	5208	5371	5534	5697	586	6023	6186	6349	163
267	42-	6511	6674	6836	6999	7161	7324	7486	7648	7811	7973	162
268	42-	8135	8297	8459	8621	8783	8944	9106	9268	9429	9591	162
269	42-	9752	9914	—	—	—	—	—	—	—	—	162
269	43-	—	—	0075	0236	0398	0559	072	0881	1042	1203	161
270	43-	1364	1525	1685	1846	2007	2167	2328	2488	2649	2809	161
271	43-	2969	313	329	345	361	377	393	409	4249	4409	160
272	43-	4569	4729	4888	5048	5207	5367	5526	5685	5844	6004	159
273	43-	6163	6322	6481	664	6799	6957	7116	7275	7433	7592	159
274	43-	7751	7909	8067	8226	8384	8542	8701	8859	9017	9175	158
275	43-	9333	9491	9648	9806	9964	—	—	—	—	—	158
275	44-	—	—	—	—	—	0122	0279	0437	0594	0752	158
276	44-	0909	1066	1224	1381	1538	1695	1852	2009	2166	2323	157
277	44-	248	2637	2793	295	3106	3263	3419	3576	3732	3889	157
278	44-	4045	4201	4357	4513	4669	4825	4981	5137	5293	5449	156
279	44-	5604	576	5915	6071	6226	6382	6537	6692	6848	7003	155
280	44-	7158	7313	7468	7623	7778	7933	8088	8242	8397	8552	155
281	44-	8706	8861	9015	917	9324	9478	9633	9787	9941	—	154
281	45-	—	—	—	—	—	—	—	—	—	0095	154
282	45-	0249	0403	0557	0711	0865	1018	1172	1326	1479	1633	154
283	45-	1786	194	2093	2247	24	2553	2706	2859	3012	3165	153
284	45-	3318	3471	3624	3777	393	4082	4235	4387	454	4692	153
285	45-	4845	4997	515	5302	5454	5606	5758	591	6062	6214	152
286	45-	6366	6518	667	6821	6973	7125	7276	7428	7579	7731	152
287	45-	7882	8033	8184	8336	8487	8638	8789	894	9091	9242	151
288	45-	9392	9543	9694	9845	9995	—	—	—	—	—	151
288	46-	—	—	—	—	—	0146	0296	0447	0597	0748	151
289	46-	0898	1048	1198	1348	1499	1649	1799	1948	2098	2248	150
No.		0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.	0	1	2	3	4	5	6	7	8	9	D
290	46- 2398	2548	2697	2847	2997	3146	3296	3445	3594	3744	150
291	46- 3893	4042	4191	434	449	4639	4788	4936	5085	5234	149
292	46- 5383	5532	568	5829	5977	6126	6274	6423	6571	6719	149
293	46- 6868	7016	7164	7312	746	7608	7756	7904	8052	82	148
294	46- 8347	8495	8643	879	8938	9085	9233	938	9527	9675	148
295	46- 9822	9969	—	—	—	—	—	—	—	—	147
295	47- —	—	0116	0263	041	0557	0704	0851	0998	1145	147
296	47- 1292	1438	1585	1732	1878	2025	2171	2318	2464	261	146
297	47- 2756	2903	3049	3195	3341	3487	3633	3779	3925	4071	146
298	47- 4216	4362	4508	4653	4799	4944	509	5235	5381	5526	146
299	47- 5671	5816	5962	6107	6252	6397	6542	6687	6832	6976	145
300	47- 7121	7266	7411	7555	77	7844	7989	8133	8278	8422	145
301	47- 8566	8711	8855	8999	9143	9287	9431	9575	9719	9863	144
302	48- 0007	0151	0294	0438	0582	0725	0869	1012	1156	1299	144
303	48- 1443	1586	1729	1872	2016	2159	2302	2445	2588	2731	143
304	48- 2874	3016	3159	3302	3445	3587	373	3872	4015	4157	143
305	48- 43	4442	4585	4727	4869	5011	5153	5295	5437	5579	142
306	48- 5721	5863	6005	6147	6289	643	6572	6714	6855	6997	142
307	48- 7138	728	7421	7563	7704	7845	7986	8127	8269	841	141
308	48- 8551	8692	8833	8974	9114	9255	9396	9537	9677	9818	141
309	48- 9958	—	—	—	—	—	—	—	—	—	140
309	49- —	0099	0239	038	052	0661	0801	0941	1081	1222	140
310	49- 1362	1502	1642	1782	1922	2062	2201	2341	2481	2621	140
311	49- 276	29	304	3179	3319	3458	3597	3737	3876	4015	139
312	49- 4155	4294	4433	4572	4711	485	4989	5128	5267	5406	139
313	49- 5544	5683	5822	596	6099	6238	6376	6515	6653	6791	139
314	49- 693	7068	7206	7344	7483	7621	7759	7897	8035	8173	138
315	49- 8311	8448	8586	8724	8862	8999	9137	9275	9412	955	138
316	49- 9687	9824	9962	—	—	—	—	—	—	—	137
316	50- —	—	—	0099	0236	0374	0511	0648	0785	0922	137
317	50- 1059	1196	1333	147	1607	1744	188	2017	2154	2291	137
318	50- 2427	2564	27	2837	2973	3109	3246	3382	3518	3655	136
319	50- 3791	3927	4063	4199	4335	4471	4607	4743	4878	5014	136
320	50- 515	5286	5421	5557	5693	5828	5964	6099	6234	637	136
321	50- 6505	664	6776	6911	7046	7181	7316	7451	7586	7721	135
322	50- 7856	7991	8126	826	8395	853	8664	8799	8934	9068	135
323	50- 9203	9337	9471	9606	974	9874	—	—	—	—	134
323	51- —	—	—	—	—	—	0009	0143	0277	0411	134
324	51- 0545	0679	0813	0947	1081	1215	1349	1482	1616	175	134
325	51- 1883	2017	2151	2284	2418	2551	2684	2818	2951	3084	133
326	51- 3218	3351	3484	3617	375	3883	4016	4149	4282	4415	133
327	51- 4548	4681	4813	4946	5079	5211	5344	5476	5609	5741	133
328	51- 5874	6006	6139	6271	6403	6535	6668	68	6932	7064	132
329	51- 7196	7328	746	7592	7724	7855	7987	8119	8251	8382	132
330	51- 8514	8646	8777	8909	904	9171	9303	9434	9566	9697	131
331	51- 9828	9959	—	—	—	—	—	—	—	—	131
331	52- —	—	009	0221	0353	0484	0615	0745	0876	1007	131
332	52- 1138	1269	14	153	1661	1792	1922	2053	2183	2314	131
333	52- 2444	2575	2705	2835	2966	3096	3226	3356	3486	3616	130
334	52- 3746	3876	4006	4136	4266	4396	4526	4656	4785	4915	130
No.	0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.	0	1	2	3	4	5	6	7	8	9	D
335	52- 5045	5174	5304	5434	5563	5693	5822	5951	6081	621	129
336	52- 6339	6469	6598	6727	6856	6985	7114	7243	7372	7501	129
337	52- 763	7759	7888	8016	8145	8274	8402	8531	866	8788	129
338	52- 8917	9045	9174	9302	943	9559	9687	9815	9943	—	128
338	53- —	—	—	—	—	—	—	—	—	0072	128
339	53- 02	0328	0456	0584	0712	084	0968	1096	1223	1351	128
340	53- 1479	1607	1734	1862	199	2117	2245	2372	25	2627	128
341	53- 2754	2882	3009	3136	3264	3391	3518	3645	3772	3899	127
342	53- 4026	4153	428	4407	4534	4661	4787	4914	5041	5167	127
343	53- 5294	5421	5547	5674	58	5927	6053	618	6306	6432	126
344	53- 6558	6685	6811	6937	7063	7189	7315	7441	7567	7693	126
345	53- 7819	7945	8071	8197	8322	8448	8574	8699	8825	8951	126
346	53- 9076	9202	9327	9452	9578	9703	9829	9954	—	—	126
346	54- —	—	—	—	—	—	—	—	0079	0204	125
347	54- 0329	0455	058	0705	083	0955	108	1205	133	1454	125
348	54- 1579	1704	1829	1953	2078	2203	2327	2452	2576	2701	125
349	54- 2825	295	3074	3199	3323	3447	3571	3696	382	3944	124
350	54- 4068	4192	4316	444	4564	4688	4812	4936	506	5183	124
351	54- 5307	5431	5555	5678	5802	5925	6049	6172	6296	6419	124
352	54- 6543	6666	6789	6913	7036	7159	7282	7405	7529	7652	123
353	54- 7775	7898	8021	8144	8267	8389	8512	8635	8758	8881	123
354	54- 9003	9126	9249	9371	9494	9616	9739	9861	9984	—	123
354	55- —	—	—	—	—	—	—	—	—	0106	123
355	55- 0228	0351	0473	0595	0717	084	0962	1084	1206	1328	122
356	55- 145	1572	1694	1816	1938	206	2181	2303	2425	2547	122
357	55- 2668	279	2911	3033	3155	3276	3398	3519	364	3762	121
358	55- 3883	4004	4126	4247	4368	4489	461	4731	4852	4973	121
359	55- 5094	5215	5336	5457	5578	5699	582	594	6061	6182	121
360	55- 6303	6423	6544	6664	6785	6905	7026	7146	7267	7387	120
361	55- 7507	7627	7748	7868	7988	8108	8228	8349	8469	8589	120
362	55- 8709	8829	8948	9068	9188	9308	9428	9548	9667	9787	120
363	55- 9907	—	—	—	—	—	—	—	—	—	120
363	56- —	0026	0146	0265	0385	0504	0624	0743	0863	0982	119
364	56- 1101	1221	134	1459	1578	1698	1817	1936	2055	2174	119
365	56- 2293	2412	2531	265	2769	2887	3006	3125	3244	3362	119
366	56- 3481	36	3718	3837	3955	4074	4192	4311	4429	4548	119
367	56- 4666	4784	4903	5021	5139	5257	5376	5494	5612	573	118
368	56- 5848	5966	6084	6202	632	6437	6555	6673	6791	6909	118
369	56- 7026	7144	7262	7379	7497	7614	7732	7849	7967	8084	118
370	56- 8202	8319	8436	8554	8671	8788	8905	9023	914	9257	117
371	56- 9374	9491	9608	9725	9842	9959	—	—	—	—	117
371	57- —	—	—	—	—	—	0076	0193	0309	0426	117
372	57- 0543	066	0776	0893	101	1126	1243	1359	1476	1592	117
373	57- 1709	1825	1942	2058	2174	2291	2407	2523	2639	2755	116
374	57- 2872	2988	3104	322	3336	3452	3568	3684	38	3915	116
375	57- 4031	4147	4263	4379	4494	461	4726	4841	4957	5072	116
376	57- 5188	5303	5419	5534	565	5765	588	5996	6111	6226	115
377	57- 6341	6457	6572	6687	6802	6917	7032	7147	7262	7377	115
378	57- 7492	7607	7722	7836	7951	8066	8181	8295	841	8525	115
379	57- 8639	8754	8868	8983	9097	9212	9326	9441	9555	9669	114
No.	0	1	2	3	4	5	6	7	8	9	D

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No.		0	1	2	3	4	5	6	7	8	9	D
380	57-	9784	9898	—	—	—	—	—	—	—	—	114
380	58-	—	—	0012	0126	0241	0355	0469	0583	0697	0811	114
381	58-	0925	1039	1153	1267	1381	1495	1608	1722	1836	195	114
382	58-	2063	2177	2291	2404	2518	2631	2745	2858	2972	3085	114
383	58-	3199	3312	3426	3539	3652	3765	3879	3992	4105	4218	113
384	58-	4331	4444	4557	467	4783	4896	5009	5122	5235	5348	113
385	58-	5461	5574	5686	5799	5912	6024	6137	625	6362	6475	113
386	58-	6587	67	6812	6925	7037	7149	7262	7374	7486	7599	112
387	58-	7711	7823	7935	8047	816	8272	8384	8496	8608	872	112
388	58-	8832	8944	9056	9167	9279	9391	9503	9615	9726	9838	112
389	58-	995	—	—	—	—	—	—	—	—	—	112
389	59-	—	0061	0173	0284	0396	0507	0619	073	0842	0953	112
390	59-	1065	1176	1287	1399	151	1621	1732	1843	1955	2066	111
391	59-	2177	2288	2399	251	2621	2732	2843	2954	3064	3175	111
392	59-	3286	3397	3508	3618	3729	384	395	4061	4171	4282	111
393	59-	4393	4503	4614	4724	4834	4945	5055	5165	5276	5386	110
394	59-	5496	5606	5717	5827	5937	6047	6157	6267	6377	6487	110
395	59-	6597	6707	6817	6927	7037	7146	7256	7366	7476	7586	110
396	59-	7695	7805	7914	8024	8134	8243	8353	8462	8572	8681	110
397	59-	8791	89	9009	9119	9228	9337	9446	9556	9665	9774	109
398	59-	9883	9992	—	—	—	—	—	—	—	—	109
398	60-	—	—	0101	021	0319	0428	0537	0646	0755	0864	109
399	60-	0973	1082	1191	1299	1408	1517	1625	1734	1843	1951	109
400	60-	206	2169	2277	2386	2494	2603	2711	2819	2928	3036	108
401	60-	3144	3253	3361	3469	3577	3686	3794	3902	401	4118	108
402	60-	4226	4334	4442	455	4658	4766	4874	4982	5089	5197	108
403	60-	5305	5413	5521	5628	5736	5844	5951	6059	6166	6274	108
404	60-	6381	6489	6596	6704	6811	6919	7026	7133	7241	7348	107
405	60-	7455	7562	7669	7777	7884	7991	8098	8205	8312	8419	107
406	60-	8526	8633	874	8847	8954	9061	9167	9274	9381	9488	107
407	60-	9594	9701	9808	9914	—	—	—	—	—	—	107
407	61-	—	—	—	—	0021	0128	0234	0341	0447	0554	107
408	61-	066	0767	0873	0979	1086	1192	1298	1405	1511	1617	106
409	61-	1723	1829	1936	2042	2148	2254	236	2466	2572	2678	106
410	61-	2784	289	2996	3102	3207	3313	3419	3525	363	3736	106
411	61-	3842	3947	4053	4159	4264	437	4475	4581	4686	4792	106
412	61-	4897	5003	5108	5213	5319	5424	5529	5634	574	5845	105
413	61-	595	6055	616	6265	637	6476	6581	6686	679	6895	105
414	61-	7	7105	721	7315	742	7525	7629	7734	7839	7943	105
415	61-	8048	8153	8257	8362	8466	8571	8676	878	8884	8989	105
416	61-	9093	9198	9302	9406	9511	9615	9719	9824	9928	—	105
416	62-	—	—	—	—	—	—	—	—	—	0032	104
417	62-	0136	024	0344	0448	0552	0656	076	0864	0968	1072	104
418	62-	1176	128	1384	1488	1592	1695	1799	1903	2007	211	104
419	62-	2214	2318	2421	2525	2628	2732	2835	2939	3042	3146	104
420	62-	3249	3353	3456	3559	3663	3766	3869	3973	4076	4179	103
421	62-	4282	4385	4488	4591	4695	4798	4901	5004	5107	521	103
422	62-	5312	5415	5518	5621	5724	5827	5929	6032	6135	6238	103
423	62-	634	6443	6546	6648	6751	6853	6956	7058	7161	7263	103
424	62-	7366	7468	7571	7673	7775	7878	798	8082	8185	8287	102
No.		0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

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No.	0	1	2	3	4	5	6	7	8	9	D
425	62- 8389	8491	8593	8695	8797	89	9002	9104	9206	9308	102
426	62- 941	9512	9613	9715	9817	9919	—	—	—	—	102
426	63- —	—	—	—	—	—	0021	0123	0224	0326	102
427	63- 0428	053	0631	0733	0835	0936	1038	1139	1241	1342	102
428	63- 1444	1545	1647	1748	1849	1951	2052	2153	2255	2356	101
429	63- 2457	2559	266	2761	2862	2963	3064	3165	3266	3367	101
430	63- 3468	3569	367	3771	3872	3973	4074	4175	4276	4376	101
431	63- 4477	4578	4679	4779	488	4981	5081	5182	5283	5383	101
432	63- 5484	5584	5685	5785	5886	5986	6087	6187	6287	6388	100
433	63- 6488	6588	6688	6789	6889	6989	7089	7189	729	739	100
434	63- 749	759	769	779	789	799	809	819	829	8389	100
435	63- 8489	8589	8689	8789	8888	8988	9088	9188	9287	9387	100
436	63- 9486	9586	9686	9785	9885	9984	—	—	—	—	100
436	64- —	—	—	—	—	—	0084	0183	0283	0382	99
437	64- 0481	0581	068	0779	0879	0978	1077	1177	1276	1375	99
438	64- 1474	1573	1672	1771	1871	197	2069	2168	2267	2366	99
439	64- 2465	2563	2662	2761	286	2959	3058	3156	3255	3354	99
440	64- 3453	3551	365	3749	3847	3946	4044	4143	4242	434	99
441	64- 4439	4537	4636	4734	4832	4931	5029	5127	5226	5324	98
442	64- 5422	5521	5619	5717	5815	5913	6011	611	6208	6306	98
443	64- 6404	6502	66	6698	6796	6894	6992	7089	7187	7285	98
444	64- 7383	7481	7579	7676	7774	7872	7969	8067	8165	8262	98
445	64- 836	8458	8555	8653	875	8848	8945	9043	914	9237	97
446	64- 9335	9432	953	9627	9724	9821	9919	—	—	—	97
446	65- —	—	—	—	—	—	—	0016	0113	021	97
447	65- 0308	0405	0502	0599	0696	0793	089	0987	1084	1181	97
448	65- 1278	1375	1472	1569	1666	1762	1859	1956	2053	215	97
449	65- 2246	2343	244	2536	2633	273	2826	2923	3019	3116	97
450	65- 3213	3309	3405	3502	3598	3695	3791	3888	3984	408	96
451	65- 4177	4273	4369	4465	4562	4658	4754	485	4946	5042	96
452	65- 5138	5235	5331	5427	5523	5619	5715	581	5906	6002	96
453	65- 6098	6194	629	6386	6482	6577	6673	6769	6864	696	96
454	65- 7056	7152	7247	7343	7438	7534	7629	7725	782	7916	96
455	65- 8011	8107	8202	8298	8393	8488	8584	8679	8774	887	95
456	65- 8965	906	9155	925	9346	9441	9536	9631	9726	9821	95
457	65- 9916	—	—	—	—	—	—	—	—	—	95
457	66- —	0011	0106	0201	0296	0391	0486	0581	0676	0771	95
458	66- 0865	096	1055	115	1245	1339	1434	1529	1623	1718	95
459	66- 1813	1907	2002	2096	2191	2286	238	2475	2569	2663	95
460	66- 2758	2852	2947	3041	3135	323	3324	3418	3512	3607	94
461	66- 3701	3795	3889	3983	4078	4172	4266	436	4454	4548	94
462	66- 4642	4736	483	4924	5018	5112	5206	5299	5393	5487	94
463	66- 5581	5675	5769	5862	5956	605	6143	6237	6331	6424	94
464	66- 6518	6612	6705	6799	6892	6986	7079	7173	7266	736	94
465	66- 7453	7546	764	7733	7826	792	8013	8106	8199	8293	93
466	66- 8386	8479	8572	8665	8759	8852	8945	9038	9131	9224	93
467	66- 9317	941	9503	9596	9689	9782	9875	9967	—	—	93
467	67- —	—	—	—	—	—	—	—	006	0153	93
468	67- 0246	0339	0431	0524	0617	071	0802	0895	0988	108	93
469	67- 1173	1265	1358	1451	1543	1636	1728	1821	1913	2005	93
No.	0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.	0	1	2	3	4	5	6	7	8	9	D
470	67- 2098	219	2283	2375	2467	256	2652	2744	2836	2929	92
471	67- 3021	3113	3205	3297	339	3482	3574	3666	3758	385	92
472	67- 3942	4034	4126	4218	431	4402	4494	4586	4677	4769	92
473	67- 4861	4953	5045	5137	5228	532	5412	5503	5595	5687	92
474	67- 5778	587	5962	6053	6145	6236	6328	6419	6511	6602	92
475	67- 6694	6785	6876	6968	7059	7151	7242	7333	7424	7516	91
476	67- 7607	7698	7789	7881	7972	8063	8154	8245	8336	8427	91
477	67- 8518	8609	87	8791	8882	8973	9064	9155	9246	9337	91
478	67- 9428	9519	961	97	9791	9882	9973	—	—	—	91
478	68- —	—	—	—	—	—	—	0063	0154	0245	91
479	68- 0336	0426	0517	0607	0698	0789	0879	097	106	1151	91
480	68- 1241	1332	1422	1513	1603	1693	1784	1874	1964	2055	90
481	68- 2145	2235	2326	2416	2506	2596	2686	2777	2867	2957	90
482	68- 3047	3137	3227	3317	3407	3497	3587	3677	3767	3857	90
483	68- 3947	4037	4127	4217	4307	4396	4486	4576	4666	4756	90
484	68- 4845	4935	5025	5114	5204	5294	5383	5473	5563	5652	90
485	68- 5742	5831	5921	601	61	6189	6279	6368	6458	6547	89
486	68- 6636	6726	6815	6904	6994	7083	7172	7261	7351	744	89
487	68- 7529	7618	7707	7796	7886	7975	8064	8153	8242	8331	89
488	68- 842	8509	8598	8687	8776	8865	8953	9042	9131	922	89
489	68- 9309	9398	9486	9575	9664	9753	9841	993	—	—	89
489	69- —	—	—	—	—	—	—	—	0019	0107	89
490	69- 0196	0285	0373	0462	055	0639	0728	0816	0905	0993	89
491	69- 1081	117	1258	1347	1435	1524	1612	17	1789	1877	88
492	69- 1965	2053	2142	223	2318	2406	2494	2583	2671	2759	88
493	69- 2847	2935	3023	3111	3199	3287	3375	3463	3551	3639	88
494	69- 3727	3815	3903	3991	4078	4166	4254	4342	443	4517	88
495	69- 4605	4693	4781	4868	4956	5044	5131	5219	5307	5394	88
496	69- 5482	5569	5657	5744	5832	5919	6007	6094	6182	6269	87
497	69- 6356	6444	6531	6618	6706	6793	688	6968	7055	7142	87
498	69- 7229	7317	7404	7491	7578	7665	7752	7839	7926	8014	87
499	69- 8101	8188	8275	8362	8449	8535	8622	8709	8796	8883	87
500	69- 897	9057	9144	9231	9317	9404	9491	9578	9664	9751	87
501	69- 9838	9924	—	—	—	—	—	—	—	—	87
501	70- —	—	0011	0098	0184	0271	0358	0444	0531	0617	87
502	70- 0704	079	0877	0963	105	1136	1222	1309	1395	1482	86
503	70- 1568	1654	1741	1827	1913	1999	2086	2172	2258	2344	86
504	70- 2431	2517	2603	2689	2775	2861	2947	3033	3119	3205	86
505	70- 3291	3377	3463	3549	3635	3721	3807	3893	3979	4065	86
506	70- 4151	4236	4322	4408	4494	4579	4665	4751	4837	4922	86
507	70- 5008	5094	5179	5265	535	5436	5522	5607	5693	5778	86
508	70- 5864	5949	6035	612	6206	6291	6376	6462	6547	6632	85
509	70- 6718	6803	6888	6974	7059	7144	7229	7315	74	7485	85
510	70- 757	7655	774	7826	7911	7996	8081	8166	8251	8336	85
511	70- 8421	8506	8591	8676	8761	8846	8931	9015	91	9185	85
512	70- 927	9355	944	9524	9609	9694	9779	9863	9948	—	85
512	71- —	—	—	—	—	—	—	—	—	0033	85
513	71- 0117	0202	0287	0371	0456	054	0625	071	0794	0879	85
514	71- 0963	1048	1132	1217	1301	1385	147	1554	1639	1723	84
No.	0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.	0	1	2	3	4	5	6	7	8	9	D
515	71- 1807	1892	1976	206	2144	2229	2313	2397	2481	2566	84
516	71- 265	2734	2818	2902	2986	307	3154	3238	3323	3407	84
517	71- 3491	3575	3659	3742	3826	391	3994	4078	4162	4246	84
518	71- 433	4414	4497	4581	4665	4749	4833	4916	5	5084	84
519	71- 5167	5251	5335	5418	5502	5586	5669	5753	5836	592	84
520	71- 6003	6087	617	6254	6337	6421	6504	6588	6671	6754	83
521	71- 6838	6921	7004	7088	7171	7254	7338	7421	7504	7587	83
522	71- 7671	7754	7837	792	8003	8086	8169	8253	8336	8419	83
523	71- 8502	8585	8668	8751	8834	8917	9	9083	9165	9248	83
524	71- 9331	9414	9497	958	9663	9745	9828	9911	9994	—	83
524	72- —	—	—	—	—	—	—	—	—	0077	83
525	72- 0159	0242	0325	0407	049	0573	0655	0738	0821	0903	83
526	72- 0986	1068	1151	1233	1316	1398	1481	1563	1646	1728	82
527	72- 1811	1893	1975	2058	214	2222	2305	2387	2469	2552	82
528	72- 2634	2716	2798	2881	2963	3045	3127	3209	3291	3374	82
529	72- 3456	3538	362	3702	3784	3866	3948	403	4112	4194	82
530	72- 4276	4358	444	4522	4604	4685	4767	4849	4931	5013	82
531	72- 5095	5176	5258	534	5422	5503	5585	5667	5748	583	82
532	72- 5912	5993	6075	6156	6238	632	6401	6483	6564	6646	82
533	72- 6727	6809	689	6972	7053	7134	7216	7297	7379	746	81
534	72- 7541	7623	7704	7785	7866	7948	8029	811	8191	8273	81
535	72- 8354	8435	8516	8597	8678	8759	8841	8922	9003	9084	81
536	72- 9165	9246	9327	9408	9489	957	9651	9732	9813	9893	81
537	72- 9974	—	—	—	—	—	—	—	—	—	81
537	73- —	0055	0136	0217	0298	0378	0459	054	0621	0702	81
538	73- 0782	0863	0944	1024	1105	1186	1266	1347	1428	1508	81
539	73- 1589	1669	175	183	1911	1991	2072	2152	2233	2313	81
540	73- 2394	2474	2555	2635	2715	2796	2876	2956	3037	3117	80
541	73- 3197	3278	3358	3438	3518	3598	3679	3759	3839	3919	80
542	73- 3999	4079	416	424	432	44	448	456	464	472	80
543	73- 48	488	496	504	512	52	5279	5359	5439	5519	80
544	73- 5599	5679	5759	5838	5918	5998	6078	6157	6237	6317	80
545	73- 6397	6476	6556	6635	6715	6795	6874	6954	7034	7113	80
546	73- 7193	7272	7352	7431	7511	759	767	7749	7829	7908	79
547	73- 7987	8067	8146	8225	8305	8384	8463	8543	8622	8701	79
548	73- 8781	886	8939	9018	9097	9177	9256	9335	9414	9493	79
549	73- 9572	9651	9731	981	9889	9968	—	—	—	—	79
549	74- —	—	—	—	—	—	0047	0126	0205	0284	79
550	74- 0363	0442	0521	06	0678	0757	0836	0915	0994	1073	79
551	74- 1152	123	1309	1388	1467	1546	1624	1703	1782	186	79
552	74- 1939	2018	2096	2175	2254	2332	2411	2489	2568	2647	79
553	74- 2725	2804	2882	2961	3039	3118	3196	3275	3353	3431	78
554	74- 351	3588	3667	3745	3823	3902	398	4058	4136	4215	78
555	74- 4293	4371	4449	4528	4606	4684	4762	484	4919	4997	78
556	74- 5075	5153	5231	5309	5387	5465	5543	5621	5699	5777	78
557	74- 5855	5933	6011	6089	6167	6245	6323	6401	6479	6556	78
558	74- 6634	6712	679	6868	6945	7023	7101	7179	7256	7334	78
559	74- 7412	7489	7567	7645	7722	78	7878	7955	8033	811	78
No.	0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.		0	1	2	3	4	5	6	7	8	9	D
560	74-	8188	8266	8343	8421	8498	8576	8653	8731	8808	8885	77
561	74-	8963	904	9118	9195	9272	935	9427	9504	9582	9659	77
562	74-	9736	9814	9891	9968	—	—	—	—	—	—	77
562	75-	—	—	—	—	0045	0123	02	0277	0354	0431	77
563	75-	0508	0586	0663	074	0817	0894	0971	1048	1125	1202	77
564	75-	1279	1356	1433	151	1587	1664	1741	1818	1895	1972	77
565	75-	2048	2125	2202	2279	2356	2433	2509	2586	2663	274	77
566	75-	2816	2893	297	3047	3123	32	3277	3353	343	3506	77
567	75-	3583	366	3736	3813	3889	3966	4042	4119	4195	4272	77
568	75-	4348	4425	4501	4578	4654	473	4807	4883	496	5036	76
569	75-	5112	5189	5265	5341	5417	5494	557	5646	5722	5799	76
570	75-	5875	5951	6027	6103	618	6256	6332	6408	6484	656	76
571	75-	6636	6712	6788	6864	694	7016	7092	7168	7244	732	76
572	75-	7396	7472	7548	7624	77	7775	7851	7927	8003	8079	76
573	75-	8155	823	8306	8382	8458	8533	8609	8685	8761	8836	76
574	75-	8912	8988	9063	9139	9214	929	9366	9441	9517	9592	76
575	75-	9668	9743	9819	9894	997	—	—	—	—	—	76
575	76-	—	—	—	—	—	0045	0121	0196	0272	0347	75
576	76-	0422	0498	0573	0649	0724	0799	0875	095	1025	1101	75
577	76-	1176	1251	1326	1402	1477	1552	1627	1702	1778	1853	75
578	76-	1928	2003	2078	2153	2228	2303	2378	2453	2529	2604	75
579	76-	2679	2754	2829	2904	2978	3053	3128	3203	3278	3353	75
580	76-	3428	3503	3578	3653	3727	3802	3877	3952	4027	4101	75
581	76-	4176	4251	4326	44	4475	455	4624	4699	4774	4848	75
582	76-	4923	4998	5072	5147	5221	5296	537	5445	552	5594	75
583	76-	5669	5743	5818	5892	5966	6041	6115	619	6264	6338	74
584	76-	6413	6487	6562	6636	671	6785	6859	6933	7007	7082	74
585	76-	7156	723	7304	7379	7453	7527	7601	7675	7749	7823	74
586	76-	7898	7972	8046	812	8194	8268	8342	8416	849	8564	74
587	76-	8638	8712	8786	886	8934	9008	9082	9156	923	9303	74
588	76-	9377	9451	9525	9599	9673	9746	982	9894	9968	—	74
588	77-	—	—	—	—	—	—	—	—	—	0042	74
589	77-	0115	0189	0263	0336	041	0484	0557	0631	0705	0778	74
590	77-	0852	0926	0999	1073	1146	122	1293	1367	144	1514	74
591	77-	1587	1661	1734	1808	1881	1955	2028	2102	2175	2248	73
592	77-	2322	2395	2468	2542	2615	2688	2762	2835	2908	2981	73
593	77-	3055	3128	3201	3274	3348	3421	3494	3567	364	3713	73
594	77-	3786	386	3933	4006	4079	4152	4225	4298	4371	4444	73
595	77-	4517	459	4663	4736	4809	4882	4955	5028	51	5173	73
596	77-	5246	5319	5392	5465	5538	561	5683	5756	5829	5902	73
597	77-	5974	6047	612	6193	6265	6338	6411	6483	6556	6629	73
598	77-	6701	6774	6846	6919	6992	7064	7137	7209	7282	7354	73
599	77-	7427	7499	7572	7644	7717	7789	7862	7934	8006	8079	72
600	77-	8151	8224	8296	8368	8441	8513	8585	8658	873	8802	72
601	77-	8874	8947	9019	9091	9163	9236	9308	938	9452	9524	72
602	77-	9596	9669	9741	9813	9885	9957	—	—	—	—	72
602	78-	—	—	—	—	—	—	0029	0101	0173	0245	72
603	78-	0317	0389	0461	0533	0605	0677	0749	0821	0893	0965	72
604	78-	1037	1109	1181	1253	1324	1396	1468	154	1612	1684	72
No		0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.	0	1	2	3	4	5	6	7	8	9	D
605	78- 1755	1827	1899	1971	2042	2114	2186	2258	2329	2401	72
606	78- 2473	2544	2616	2688	2759	2831	2902	2974	3046	3117	72
607	78- 3189	326	3332	3403	3475	3546	3618	3689	3761	3832	71
608	78- 3904	3975	4046	4118	4189	4261	4332	4403	4475	4546	71
609	78- 4617	4689	476	4831	4902	4974	5045	5116	5187	5259	71
610	78- 533	5401	5472	5543	5615	5686	5757	5828	5899	597	71
611	78- 6041	6112	6183	6254	6325	6396	6467	6538	6609	668	71
612	78- 6751	6822	6893	6964	7035	7106	7177	7248	7319	739	71
613	78- 746	7531	7602	7673	7744	7815	7885	7956	8027	8098	71
614	78- 8168	8239	831	8381	8451	8522	8593	8663	8734	8804	71
615	78- 8875	8946	9016	9087	9157	9228	9299	9369	944	951	71
616	78- 9581	9651	9722	9792	9863	9933	—	—	—	—	70
616	79- —	—	—	—	—	—	0004	0074	0144	0215	70
617	79- 0285	0356	0426	0496	0567	0637	0707	0778	0848	0918	70
618	79- 0988	1059	1129	1199	1269	134	141	148	155	162	70
619	79- 1691	1761	1831	1901	1971	2041	2111	2181	2252	2322	70
620	79- 2392	2462	2532	2602	2672	2742	2812	2882	2952	3022	70
621	79- 3092	3162	3231	3301	3371	3441	3511	3581	3651	3721	70
622	79- 379	386	393	4	407	4139	4209	4279	4349	4418	70
623	79- 4488	4558	4627	4697	4767	4836	4906	4976	5045	5115	70
624	79- 5185	5254	5324	5393	5463	5532	5602	5672	5741	5811	70
625	79- 588	5949	6019	6088	6158	6227	6297	6366	6436	6505	69
626	79- 6574	6644	6713	6782	6852	6921	699	706	7129	7198	69
627	79- 7268	7337	7406	7475	7545	7614	7683	7752	7821	789	69
628	79- 796	8029	8098	8167	8236	8305	8374	8443	8513	8582	69
629	79- 8651	872	8789	8858	8927	8996	9065	9134	9203	9272	69
630	79- 9341	9409	9478	9547	9616	9685	9754	9823	9892	9961	69
631	80- 0029	0098	0167	0236	0305	0373	0442	0511	058	0648	69
632	80- 0717	0786	0854	0923	0992	1061	1129	1198	1266	1335	69
633	80- 1404	1472	1541	1609	1678	1747	1815	1884	1952	2021	69
634	80- 2089	2158	2226	2295	2363	2432	25	2568	2637	2705	69
635	80- 2774	2842	291	2979	3047	3116	3184	3252	3321	3389	68
636	80- 3457	3525	3594	3662	373	3798	3867	3935	4003	4071	68
637	80- 4139	4208	4276	4344	4412	448	4548	4616	4685	4753	68
638	80- 4821	4889	4957	5025	5093	5161	5229	5297	5365	5433	68
639	80- 5501	5569	5637	5705	5773	5841	5908	5976	6044	6112	68
640	80- 618	6248	6316	6384	6451	6519	6587	6655	6723	679	68
641	80- 6858	6926	6994	7061	7129	7197	7264	7332	74	7467	68
642	80- 7535	7603	767	7738	7806	7873	7941	8008	8076	8143	68
643	80- 8211	8279	8346	8414	8481	8549	8616	8684	8751	8818	67
644	80- 8886	8953	9021	9088	9156	9223	929	9358	9425	9492	67
645	80- 956	9627	9694	9762	9829	9896	9964	—	—	—	67
645	81- —	—	—	—	—	—	—	0031	0098	0165	67
646	81- 0233	03	0367	0434	0501	0569	0636	0703	077	0837	67
647	81- 0904	0971	1039	1106	1173	124	1307	1374	1441	1508	67
648	81- 1575	1642	1709	1776	1843	191	1977	2044	2111	2178	67
649	81- 2245	2312	2379	2445	2512	2579	2646	2713	278	2847	67
650	81- 2913	298	3047	3114	3181	3247	3314	3381	3448	3514	67
651	81- 3581	3648	3714	3781	3848	3914	3981	4048	4114	4181	67
652	81- 4248	4314	4381	4447	4514	4581	4647	4714	478	4847	67
653	81- 4913	498	5046	5113	5179	5246	5312	5378	5445	5511	66
654	81- 5578	5644	5711	5777	5843	591	5976	6042	6109	6175	66
No.	0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.	0	1	2	3	4	5	6	7	8	9	D
655	81- 6241	6308	6374	644	6506	6573	6639	6705	6771	6838	66
656	81- 6904	697	7036	7102	7169	7235	7301	7367	7433	7499	66
657	81- 7565	7631	7698	7764	783	7896	7962	8028	8094	816	66
658	81- 8226	8292	8358	8424	849	8556	8622	8688	8754	882	66
659	81- 8885	8951	9017	9083	9149	9215	9281	9346	9412	9478	66
660	81- 9544	961	9676	9741	9807	9873	9939	—	—	—	66
660	82- —	—	—	—	—	—	—	0004	007	0136	66
661	82- 0201	0267	0333	0399	0464	053	0595	0661	0727	0792	66
662	82- 0858	0924	0989	1055	112	1186	1251	1317	1382	1448	66
663	82- 1514	1579	1645	171	1775	1841	1906	1972	2037	2103	65
664	82- 2168	2233	2299	2364	243	2495	256	2626	2691	2756	65
665	82- 2822	2887	2952	3018	3083	3148	3213	3279	3344	3409	65
666	82- 3474	3539	3605	367	3735	38	3865	393	3996	4061	65
667	82- 4126	4191	4256	4321	4386	4451	4516	4581	4646	4711	65
668	82- 4776	4841	4906	4971	5036	5101	5166	5231	5296	5361	65
669	82- 5426	5491	5556	5621	5686	5751	5815	588	5945	601	65
670	82- 6075	614	6204	6269	6334	6399	6464	6528	6593	6658	65
671	82- 6723	6787	6852	6517	6981	7046	7111	7175	724	7305	65
672	82- 7369	7434	7499	7563	7628	7692	7757	7821	7886	7951	65
673	82- 8015	808	8144	8209	8273	8338	8402	8467	8531	8595	64
674	82- 866	8724	8789	8853	8918	8982	9046	9111	9175	9239	64
675	82- 9304	9368	9432	9497	9561	9625	969	9754	9818	9882	64
676	82- 9947	—	—	—	—	—	—	—	—	—	64
676	83- —	0011	0075	0139	0204	0268	0332	0396	046	0525	64
677	83- 0589	0653	0717	0781	0845	0909	0973	1037	1102	1166	64
678	83- 123	1294	1358	1422	1486	155	1614	1678	1742	1806	64
679	83- 187	1934	1998	2062	2126	2189	2253	2317	2381	2445	64
680	83- 2509	2573	2637	27	2764	2828	2892	2956	302	3083	64
681	83- 3147	3211	3275	3338	3402	3466	353	3593	3657	3721	64
682	83- 3784	3848	3912	3975	4039	4103	4166	423	4294	4357	64
683	83- 4421	4484	4548	4611	4675	4739	4802	4866	4929	4993	64
684	83- 5056	512	5183	5247	531	5373	5437	55	5564	5627	63
685	83- 5691	5754	5817	5881	5944	6007	6071	6134	6197	6261	63
686	83- 6324	6387	6451	6514	6577	6641	6704	6767	683	6894	63
687	83- 6957	702	7083	7146	721	7273	7336	7399	7462	7525	63
688	83- 7588	7652	7715	7778	7841	7904	7967	803	8093	8156	63
689	83- 8219	8282	8345	8408	8471	8534	8597	866	8723	8786	63
690	83- 8849	8912	8975	9038	9101	9164	9227	9289	9352	9415	63
691	83- 9478	9541	9604	9667	9729	9792	9855	9918	9981	—	63
691	84- —	—	—	—	—	—	—	—	—	0043	63
692	84 0106	0169	0232	0294	0357	042	0482	0545	0608	0671	63
693	84- 0733	0796	0859	0921	0984	1046	1109	1172	1234	1297	63
694	84- 1359	1422	1485	1547	161	1672	1735	1797	186	1922	63
695	84- 1985	2047	211	2172	2235	2297	236	2422	2484	2547	62
696	84- 2609	2672	2734	2796	2859	2921	2983	3046	3108	317	62
697	84- 3233	3295	3357	342	3482	3544	3606	3669	3731	3793	62
698	84- 3855	3918	398	4042	4104	4166	4229	4291	4353	4415	62
699	84- 4477	4539	4601	4664	4726	4788	485	4912	4974	5036	62
700	84- 5098	516	5222	5284	5346	5408	547	5532	5594	5656	62
701	84- 5718	578	5842	5904	5966	6028	609	6151	6213	6275	62
702	84- 6337	6399	6461	6523	6585	6646	6708	677	6832	6894	62
703	84- 6955	7017	7079	7141	7202	7264	7326	7388	7449	7511	62
704	84- 7573	7634	7696	7758	7819	7881	7943	8004	8066	8127	62
No.	0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.		0	1	2	3	4	5	6	7	8	9	D
705	84-	8189	8251	8312	8374	8435	8497	8559	862	8682	8743	62
706	84-	8805	8866	8928	8989	9051	9112	9174	9235	9297	9358	61
707	84-	9419	9481	9542	9604	9665	9726	9788	9849	9911	9972	61
708	85-	0033	0095	0156	0217	0279	034	0401	0462	0524	0585	61
709	85-	0646	0707	0769	083	0891	0952	1014	1075	1136	1197	61
710	85-	1258	132	1381	1442	1503	1564	1625	1686	1747	1809	61
711	85-	187	1931	1992	2053	2114	2175	2236	2297	2358	2419	61
712	85-	248	2541	2602	2663	2724	2785	2846	2907	2968	3029	61
713	85-	309	315	3211	3272	3333	3394	3455	3516	3577	3637	61
714	85-	3698	3759	382	3881	3941	4002	4063	4124	4185	4245	61
715	85-	4306	4367	4428	4488	4549	461	467	4731	4792	4852	61
716	85-	4913	4974	5034	5095	5156	5216	5277	5337	5398	5459	61
717	85-	5519	558	564	5701	5761	5822	5882	5943	6003	6064	61
718	85-	6124	6185	6245	6306	6366	6427	6487	6548	6608	6668	60
719	85-	6729	6789	685	691	697	7031	7091	7152	7212	7272	60
720	85-	7332	7393	7453	7513	7574	7634	7694	7755	7815	7875	60
721	85-	7935	7995	8056	8116	8176	8236	8297	8357	8417	8477	60
722	85-	8537	8597	8657	8718	8778	8838	8898	8958	9018	9078	60
723	85-	9138	9198	9258	9318	9379	9439	9499	9559	9619	9679	60
724	85-	9739	9799	9859	9918	9978	—	—	—	—	—	60
724	86-	—	—	—	—	—	0038	0098	0158	0218	0278	60
725	86-	0338	0398	0458	0518	0578	0637	0697	0757	0817	0877	60
726	86-	0937	0996	1056	1116	1176	1236	1295	1355	1415	1475	60
727	86-	1534	1594	1654	1714	1773	1833	1893	1952	2012	2072	60
728	86-	2131	2191	2251	231	237	243	2489	2549	2608	2668	60
729	86-	2728	2787	2847	2906	2966	3025	3085	3144	3204	3263	60
730	86-	3323	3382	3442	3501	3561	362	368	3739	3799	3858	59
731	86-	3917	3977	4036	4096	4155	4214	4274	4333	4392	4452	59
732	86-	4511	457	463	4689	4748	4808	4867	4926	4985	5045	59
733	86-	5104	5163	5222	5282	5341	54	5459	5519	5578	5637	59
734	86-	5696	5755	5814	5874	5933	5992	6051	611	6169	6228	59
735	86-	6287	6346	6405	6465	6524	6583	6642	6701	676	6819	59
736	86-	6878	6937	6996	7055	7114	7173	7232	7291	735	7409	59
737	86-	7467	7526	7585	7644	7703	7762	7821	788	7939	7998	59
738	86-	8056	8115	8174	8233	8292	835	8409	8468	8527	8586	59
739	86-	8644	8703	8762	8821	8879	8938	8997	9056	9114	9173	59
740	86-	9232	929	9349	9408	9466	9525	9584	9642	9701	976	59
741	86-	9818	9877	9935	9994	—	—	—	—	—	—	59
741	87-	—	—	—	—	0053	0111	017	0228	0287	0345	59
742	87-	0404	0462	0521	0579	0638	0696	0755	0813	0872	093	58
743	87-	0989	1047	1106	1164	1223	1281	1339	1398	1456	1515	58
744	87-	1573	1631	169	1748	1806	1865	1923	1981	204	2098	58
745	87-	2156	2215	2273	2331	2389	2448	2506	2564	2622	2681	58
746	87-	2739	2797	2855	2913	2972	303	3088	3146	3204	3262	58
747	87-	3321	3379	3437	3495	3553	3611	3669	3727	3785	3844	58
748	87-	3902	396	4018	4076	4134	4192	425	4308	4366	4424	58
749	87-	4482	454	4598	4656	4714	4772	483	4888	4945	5003	58
750	87-	5061	5119	5177	5235	5293	5351	5409	5466	5524	5582	58
751	87-	564	5698	5756	5813	5871	5929	5987	6045	6102	616	58
752	87-	6218	6276	6333	6391	6449	6507	6564	6622	668	6737	58
753	87-	6795	6853	691	6968	7026	7083	7141	7199	7256	7314	58
754	87-	7371	7429	7487	7544	7602	7659	7717	7774	7832	7889	58
No.		0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.		0	1	2	3	4	5	6	7	8	9	D
755	87-	7947	8004	8062	8119	8177	8234	8292	8349	8407	8464	57
756	87-	8522	8579	8637	8694	8752	8809	8866	8924	8981	9039	57
757	87-	9096	9153	9211	9268	9325	9383	944	9497	9555	9612	57
758	87-	9669	9726	9784	9841	9898	9956	—	—	—	—	57
758	88-	—	—	—	—	—	—	0013	007	0127	0185	57
759	88-	0242	0299	0356	0413	0471	0528	0585	0642	0699	0756	57
760	88-	0814	0871	0928	0985	1042	1099	1156	1213	1271	1328	57
761	88-	1385	1442	1499	1556	1613	167	1727	1784	1841	1898	57
762	88-	1955	2012	2069	2126	2183	224	2297	2354	2411	2468	57
763	88-	2525	2581	2638	2695	2752	2809	2866	2923	298	3037	57
764	88-	3093	315	3207	3264	3321	3377	3434	3491	3548	3605	57
765	88-	3661	3718	3775	3832	3888	3945	4002	4059	4115	4172	57
766	88-	4229	4285	4342	4399	4455	4512	4569	4625	4682	4739	57
767	88-	4795	4852	4909	4965	5022	5078	5135	5192	5248	5305	57
768	88-	5361	5418	5474	5531	5587	5644	57	5757	5813	587	57
769	88-	5926	5983	6039	6096	6152	6209	6265	6321	6378	6434	56
770	88-	6491	6547	6604	666	6716	6773	6829	6885	6942	6998	56
771	88-	7054	7111	7167	7223	728	7336	7392	7449	7505	7561	56
772	88-	7617	7674	773	7786	7842	7898	7955	8011	8067	8123	56
773	88-	8179	8236	8292	8348	8404	846	8516	8573	8629	8685	56
774	88-	8741	8797	8853	8909	8965	9021	9077	9134	919	9246	56
775	88-	9302	9358	9414	947	9526	9582	9638	9694	975	9806	56
776	88-	9862	9918	9974	—	—	—	—	—	—	—	56
776	89-	—	—	—	003	0086	0141	0197	0253	0309	0365	56
777	89-	0421	0477	0533	0589	0645	07	0756	0812	0868	0924	56
778	89-	098	1035	1091	1147	1203	1259	1314	137	1426	1482	56
779	89-	1537	1593	1649	1705	176	1816	1872	1928	1983	2039	56
780	89-	2095	215	2206	2262	2317	2373	2429	2484	254	2595	56
781	89-	2651	2707	2762	2818	2873	2929	2985	304	3096	3151	56
782	89-	3207	3262	3318	3373	3429	3484	354	3595	3651	3706	56
783	89-	3762	3817	3873	3928	3984	4039	4094	415	4205	4261	55
784	89-	4316	4371	4427	4482	4538	4593	4648	4704	4759	4814	55
785	89-	487	4925	498	5036	5091	5146	5201	5257	5312	5367	55
786	89-	5423	5478	5533	5588	5644	5699	5754	5809	5864	592	55
787	89-	5975	603	6085	614	6195	6251	6306	6361	6416	6471	55
788	89-	6526	6581	6636	6692	6747	6802	6857	6912	6967	7022	55
789	89-	7077	7132	7187	7242	7297	7352	7407	7462	7517	7572	55
790	89-	7627	7682	7737	7792	7847	7902	7957	8012	8067	8122	55
791	89-	8176	8231	8286	8341	8396	8451	8506	8561	8615	867	55
792	89-	8725	878	8835	889	8944	8999	9054	9109	9164	9218	55
793	89-	9273	9328	9383	9437	9492	9547	9602	9656	9711	9766	55
794	89-	9821	9875	993	9985	—	—	—	—	—	—	55
794	90-	—	—	—	—	0039	0094	0149	0203	0258	0312	55
795	90-	0367	0422	0476	0531	0586	064	0695	0749	0804	0859	55
796	90-	0913	0968	1022	1077	1131	1186	124	1295	1349	1404	55
797	90-	1458	1513	1567	1622	1676	1731	1785	184	1894	1948	54
798	90-	2003	2057	2112	2166	2221	2275	2329	2384	2438	2492	54
799	90-	2547	2601	2655	271	2764	2818	2873	2927	2981	3036	54
800	90-	309	3144	3199	3253	3307	3361	3416	347	3524	3578	54
801	90-	3633	3687	3741	3795	3849	3904	3958	4012	4066	412	54
802	90-	4174	4229	4283	4337	4391	4445	4499	4553	4607	4661	54
803	90-	4716	477	4824	4878	4932	4986	504	5094	5148	5202	54
804	90-	5256	531	5364	5418	5472	5526	558	5634	5688	5742	54
No.		0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.		0	1	2	3	4	5	6	7	8	9	D
805	90-	5796	585	5904	5958	6012	6066	6119	6173	6227	6281	54
806	90-	6335	6389	6443	6497	6551	6604	6658	6712	6766	682	54
807	90-	6874	6927	6981	7035	7089	7143	7196	725	7304	7358	54
808	90-	7411	7465	7519	7573	7626	768	7734	7787	7841	7895	54
809	90-	7949	8002	8056	811	8163	8217	827	8324	8378	8431	54
810	90-	8485	8539	8592	8646	8699	8753	8807	886	8914	8967	54
811	90-	9021	9074	9128	9181	9235	9289	9342	9396	9449	9503	54
812	90-	9556	9609	9663	9716	977	9823	9877	993	9984	—	54
812	91-	—	—	—	—	—	—	—	—	—	0037	53
813	91-	0091	0144	0197	0251	0304	0358	0411	0464	0518	0571	53
814	91-	0624	0678	0731	0784	0838	0891	0944	0998	1051	1104	53
815	91-	1158	1211	1264	1317	1371	1424	1477	153	1584	1637	53
816	91-	169	1743	1797	185	1903	1956	2009	2063	2116	2169	53
817	91-	2222	2275	2328	2381	2435	2488	2541	2594	2647	27	53
818	91-	2753	2806	2859	2913	2966	3019	3072	3125	3178	3231	53
819	91-	3284	3337	339	3443	3496	3549	3602	3655	3708	3761	53
820	91-	3814	3867	392	3973	4026	4079	4132	4184	4237	429	53
821	91-	4343	4396	4449	4502	4555	4608	466	4713	4766	4819	53
822	91-	4872	4925	4977	503	5083	5136	5189	5241	5294	5347	53
823	91-	54	5453	5505	5558	5611	5664	5716	5769	5822	5875	53
824	91-	5927	598	6033	6085	6138	6191	6243	6296	6349	6401	53
825	91-	6454	6507	6559	6612	6664	6717	677	6822	6875	6927	53
826	91-	698	7033	7085	7138	719	7243	7295	7348	74	7453	53
827	91-	7506	7558	7611	7663	7716	7768	782	7873	7925	7978	52
828	91-	803	8083	8135	8188	824	8293	8345	8397	845	8502	52
829	91-	8555	8607	8659	8712	8764	8816	8869	8921	8973	9026	52
830	91-	9078	913	9183	9235	9287	934	9392	9444	9496	9549	52
831	91-	9601	9653	9706	9758	981	9862	9914	9967	—	—	52
831	92-	—	—	—	—	—	—	—	—	0019	0071	52
832	92-	0123	0176	0228	028	0332	0384	0436	0489	0541	0593	52
833	92-	0645	0697	0749	0801	0853	0906	0958	101	1062	1114	52
834	92-	1166	1218	127	1322	1374	1426	1478	153	1582	1634	52
835	92-	1686	1738	179	1842	1894	1946	1998	205	2102	2154	52
836	92-	2206	2258	231	2362	2414	2466	2518	257	2622	2674	52
837	92-	2725	2777	2829	2881	2933	2985	3037	3089	314	3192	52
838	92-	3244	3296	3348	3399	3451	3503	3555	3607	3658	371	52
839	92-	3762	3814	3865	3917	3969	4021	4072	4124	4176	4228	52
840	92-	4279	4331	4383	4434	4486	4538	4589	4641	4693	4744	52
841	92-	4796	4848	4899	4951	5003	5054	5106	5157	5209	5261	52
842	92-	5312	5364	5415	5467	5518	557	5621	5673	5725	5776	52
843	92-	5828	5879	5931	5982	6034	6085	6137	6188	624	6291	51
844	92-	6342	6394	6445	6497	6548	66	6651	6702	6754	6805	51
845	92-	6857	6908	6959	7011	7062	7114	7165	7216	7268	7319	51
846	92-	737	7422	7473	7524	7576	7627	7678	773	7781	7832	51
847	92-	7883	7935	7986	8037	8088	814	8191	8242	8293	8345	51
848	92-	8396	8447	8498	8549	8601	8652	8703	8754	8805	8857	51
849	92-	8908	8959	901	9061	9112	9163	9215	9266	9317	9368	51
850	92-	9419	947	9521	9572	9623	9674	9725	9776	9827	9879	51
851	92-	993	9981	—	—	—	—	—	—	—	—	51
851	93-	—	—	0032	0083	0134	0185	0236	0287	0338	0389	51
852	93-	044	0491	0542	0592	0643	0694	0745	0796	0847	0898	51
853	93-	0949	1	1051	1102	1153	1203	1254	1305	1356	1407	51
854	93-	1458	1509	156	161	1661	1712	1763	1814	1865	1915	51
No.		0	1	2	3	4	5	6	7	8	9	

Tables of Common Logarithms

(Continued.)

No.		0	1	2	3	4	5	6	7	8	9	D
855	93-	1966	2017	2068	2118	2169	222	2271	2322	2372	2423	51
856	93-	2474	2524	2575	2626	2677	2727	2778	2829	2879	293	51
857	93-	2981	3031	3082	3133	3183	3234	3285	3335	3386	3437	51
858	93-	3487	3538	3589	3639	369	374	3791	3841	3892	3943	51
859	93-	3993	4044	4094	4145	4195	4246	4296	4347	4397	4448	51
860	93-	4498	4549	4599	465	47	4751	4801	4852	4902	4953	50
861	93-	5003	5054	5104	5154	5205	5255	5306	5356	5406	5457	50
862	93-	5507	5558	5608	5658	5709	5759	5809	586	591	596	50
863	93-	6011	6061	6111	6162	6212	6262	6313	6363	6413	6463	50
864	93-	6514	6564	6614	6665	6715	6765	6815	6865	6916	6966	50
865	93-	7016	7066	7117	7167	7217	7267	7317	7367	7418	7468	50
866	93-	7518	7568	7618	7668	7718	7769	7819	7869	7919	7969	50
867	93-	8019	8069	8119	8169	8219	8269	8319	837	842	847	50
868	93-	852	857	862	867	872	877	882	887	892	897	50
869	93-	902	907	912	917	922	927	932	9369	9419	9469	50
870	93-	9519	9569	9619	9669	9719	9769	9819	9869	9918	9968	50
871	94-	0018	0068	0118	0168	0218	0267	0317	0367	0417	0467	50
872	94-	0516	0566	0616	0666	0716	0765	0815	0865	0915	0964	50
873	94-	1014	1064	1114	1163	1213	1263	1313	1362	1412	1462	50
874	94-	1511	1561	1611	166	171	176	1809	1859	1909	1958	50
875	94-	2008	2058	2107	2157	2207	2256	2306	2355	2405	2455	50
876	94-	2504	2554	2603	2653	2702	2752	2801	2851	2901	295	50
877	94-	3	3049	3099	3148	3198	3247	3297	3346	3396	3445	49
878	94-	3495	3544	3593	3643	3692	3742	3791	3841	389	3939	49
879	94-	3989	4038	4088	4137	4186	4236	4285	4335	4384	4433	49
880	94-	4483	4532	4581	4631	468	4729	4779	4828	4877	4927	49
881	94-	4976	5025	5074	5124	5173	5222	5272	5321	537	5419	49
882	94-	5469	5518	5567	5616	5665	5715	5764	5813	5862	5912	49
883	94-	5961	601	6059	6108	6157	6207	6256	6305	6354	6403	49
884	94-	6452	6501	6551	66	6649	6698	6747	6796	6845	6894	49
885	94-	6943	6992	7041	709	714	7189	7238	7287	7336	7385	49
886	94-	7434	7483	7532	7581	763	7679	7728	7777	7826	7875	49
887	94-	7924	7973	8022	807	8119	8168	8217	8266	8315	8364	49
888	94-	8413	8462	8511	856	8609	8657	8706	8755	8804	8853	49
889	94-	8902	8951	8999	9048	9097	9146	9195	9244	9292	9341	49
890	94-	939	9439	9488	9536	9585	9634	9683	9731	978	9829	49
891	94-	9878	9926	9975	—	—	—	—	—	—	—	49
891	95-	—	—	—	0024	0073	0121	017	0219	0267	0316	49
892	95-	0365	0414	0462	0511	056	0608	0657	0706	0754	0803	49
893	95-	0851	09	0949	0997	1046	1095	1143	1192	124	1289	49
894	95-	1338	1386	1435	1483	1532	158	1629	1677	1726	1775	49
895	95-	1823	1872	192	1969	2017	2066	2114	2163	2211	226	48
896	95-	2308	2356	2405	2453	2502	255	2599	2647	2696	2744	48
897	95-	2792	2841	2889	2938	2986	3034	3083	3131	318	3228	48
898	95-	3276	3325	3373	3421	347	3518	3566	3615	3663	3711	48
899	95-	376	3808	3856	3905	3953	4001	4049	4098	4146	4194	48
900	95-	4243	4291	4339	4387	4435	4484	4532	458	4628	4677	48
901	95-	4725	4773	4821	4869	4918	4966	5014	5062	511	5158	48
902	95-	5207	5255	5303	5351	5399	5447	5495	5543	5592	564	48
903	95-	5688	5736	5784	5832	588	5928	5976	6024	6072	612	48
904	95-	6168	6216	6265	6313	6361	6409	6457	6505	6553	6601	48
No.		0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.	0	1	2	3	4	5	6	7	8	9	D
905	95- 6649	6697	6745	6793	684	6888	6936	6984	7032	708	48
906	95- 7128	7176	7224	7272	732	7368	7416	7464	7512	7559	48
907	95- 7607	7655	7703	7751	7799	7847	7894	7942	799	8038	48
908	95- 8086	8134	8181	8229	8277	8325	8373	8421	8468	8516	48
909	95- 8564	8612	8659	8707	8755	8803	885	8898	8946	8994	48
910	95- 9041	9089	9137	9185	9232	928	9328	9375	9423	9471	48
911	95- 9518	9566	9614	9661	9709	9757	9804	9852	99	9947	48
912	95- 9995	—	—	—	—	—	—	—	—	—	48
912	96- —	0042	009	0138	0185	0233	028	0328	0376	0423	48
913	96- 0471	0518	0566	0613	0661	0709	0756	0804	0851	0899	48
914	96- 0946	0994	1041	1089	1136	1184	1231	1279	1326	1374	47
915	96- 1421	1469	1516	1563	1611	1658	1706	1753	1801	1848	47
916	96- 1895	1943	199	2038	2085	2132	218	2227	2275	2322	47
917	96- 2369	2417	2464	2511	2559	2606	2653	2701	2748	2795	47
918	96- 2843	289	2937	2985	3032	3079	3126	3174	3221	3268	47
919	96- 3316	3363	341	3457	3504	3552	3599	3646	3693	3741	47
920	96- 3788	3835	3882	3929	3977	4024	4071	4118	4165	4212	47
921	96- 426	4307	4354	4401	4448	4495	4542	459	4637	4684	47
922	96- 4731	4778	4825	4872	4919	4966	5013	5061	5108	5155	47
923	96- 5202	5249	5296	5343	539	5437	5484	5531	5578	5625	47
924	96- 5672	5719	5766	5813	586	5907	5954	6001	6048	6095	47
925	96- 6142	6189	6236	6283	6329	6376	6423	647	6517	6564	47
926	96- 6611	6658	6705	6752	6799	6845	6892	6939	6986	7033	47
927	96- 708	7127	7173	722	7267	7314	7361	7408	7454	7501	47
928	96- 7548	7595	7642	7688	7735	7782	7829	7875	7922	7969	47
929	96- 8016	8062	8109	8156	8203	8249	8296	8343	839	8436	47
930	96- 8483	853	8576	8623	867	8716	8763	881	8856	8903	47
931	96- 895	8996	9043	909	9136	9183	9229	9276	9323	9369	47
932	96- 9416	9463	9509	9556	9602	9649	9695	9742	9789	9835	47
933	96- 9882	9928	9975	—	—	—	—	—	—	—	47
933	97- —	—	—	0021	0068	0114	0161	0207	0254	03	47
934	97- 0347	0393	044	0486	0533	0579	0626	0672	0719	0765	46
935	97- 0812	0858	0904	0951	0997	1044	109	1137	1183	1229	46
936	97- 1276	1322	1369	1415	1461	1508	1554	1601	1647	1693	46
937	97- 174	1786	1832	1879	1925	1971	2018	2064	211	2157	46
938	97- 2203	2249	2295	2342	2388	2434	2481	2527	2573	2619	46
939	97- 2666	2712	2758	2804	2851	2897	2943	2989	3035	3082	46
940	97- 3128	3174	322	3266	3313	3359	3405	3451	3497	3543	46
941	97- 359	3636	3682	3728	3774	382	3866	3913	3959	4005	46
942	97- 4051	4097	4143	4189	4235	4281	4327	4374	442	4466	46
943	97- 4512	4558	4604	465	4696	4742	4788	4834	488	4926	46
944	97- 4972	5018	5064	511	5156	5202	5248	5294	534	5386	46
945	97- 5432	5478	5524	557	5616	5662	5707	5753	5799	5845	46
946	97- 5891	5937	5983	6029	6075	6121	6167	6212	6258	6304	46
947	97- 635	6396	6442	6488	6533	6579	6625	6671	6717	6763	46
948	97- 6808	6854	69	6946	6992	7037	7083	7129	7175	722	46
949	97- 7266	7312	7358	7403	7449	7495	7541	7586	7632	7678	46
950	97- 7724	7769	7815	7861	7906	7952	7998	8043	8089	8135	46
951	97- 8181	8226	8272	8317	8363	8409	8454	85	8546	8591	46
952	97- 8637	8683	8728	8774	8819	8865	8911	8956	9002	9047	46
953	97- 9093	9138	9184	923	9275	9321	9366	9412	9457	9503	46
954	97- 9548	9594	9639	9685	973	9776	9821	9867	9912	9958	46
No.	0	1	2	3	4	5	6	7	8	9	D

Tables of Common Logarithms

(Continued.)

No.	0	1	2	3	4	5	6	7	8	9	D
955	98- 0003	0049	0094	014	0185	0231	0276	0322	0367	0412	45
956	98- 0458	0503	0549	0594	064	0685	073	0776	0821	0867	45
957	98- 0912	0957	1003	1048	1093	1139	1184	1229	1275	132	45
958	98- 1366	1411	1456	1501	1547	1592	1637	1683	1728	1773	45
959	98- 1819	1864	1909	1954	2	2045	209	2135	2181	2226	45
960	98- 2271	2316	2362	2407	2452	2497	2543	2588	2633	2678	45
961	98- 2723	2769	2814	2859	2904	2949	2994	304	3085	313	45
962	98- 3175	322	3265	331	3356	3401	3446	3491	3536	3581	45
963	98- 3626	3671	3716	3762	3807	3852	3897	3942	3987	4032	45
964	98- 4077	4122	4167	4212	4257	4302	4347	4392	4437	4482	45
965	98- 4527	4572	4617	4662	4707	4752	4797	4842	4887	4932	45
966	98- 4977	5022	5067	5112	5157	5202	5247	5292	5337	5382	45
967	98- 5426	5471	5516	5561	5606	5651	5696	5741	5786	583	45
968	98- 5875	592	5965	601	6055	61	6144	6189	6234	6279	45
969	98- 6324	6369	6413	6458	6503	6548	6593	6637	6682	6727	45
970	98- 6772	6817	6861	6906	6951	6996	704	7085	713	7175	45
971	98- 7219	7264	7309	7353	7398	7443	7488	7532	7577	7622	45
972	98- 7666	7711	7756	78	7845	789	7934	7979	8024	8068	45
973	98- 8113	8157	8202	8247	8291	8336	8381	8425	847	8514	45
974	98- 8559	8604	8648	8693	8737	8782	8826	8871	8916	896	45
975	98- 9005	9049	9094	9138	9183	9227	9272	9316	9361	9405	45
976	98- 945	9494	9539	9583	9628	9672	9717	9761	9806	985	44
977	98- 9895	9939	9983	—	—	—	—	—	—	—	44
977	99- —	—	—	0028	0072	0117	0161	0206	025	0294	44
978	99- 0339	0383	0428	0472	0516	0561	0605	065	0694	0738	44
979	99- 0783	0827	0871	0916	096	1004	1049	1093	1137	1182	44
980	99- 1226	127	1315	1359	1403	1448	1492	1536	158	1625	44
981	99- 1669	1713	1758	1802	1846	189	1935	1979	2023	2067	44
982	99- 2111	2156	22	2244	2288	2333	2377	2421	2465	2509	44
983	99- 2554	2598	2642	2686	273	2774	2819	2863	2907	2951	44
984	99- 2995	3039	3083	3127	3172	3216	326	3304	3348	3392	44
985	99- 3436	348	3524	3568	3613	3657	3701	3745	3789	3833	44
986	99- 3877	3921	3965	4009	4053	4097	4141	4185	4229	4273	44
987	99- 4317	4361	4405	4449	4493	4537	4581	4625	4669	4713	44
988	99- 4757	4801	4845	4889	4933	4977	5021	5065	5108	5152	44
989	99- 5196	524	5284	5328	5372	5416	546	5504	5547	5591	44
990	99- 5635	5679	5723	5767	5811	5854	5898	5942	5986	603	44
991	99- 6074	6117	6161	6205	6249	6293	6337	638	6424	6468	44
992	99- 6512	6555	6599	6643	6687	6731	6774	6818	6862	6906	44
993	99- 6949	6993	7037	708	7124	7168	7212	7255	7299	7343	44
994	99- 7386	743	7474	7517	7561	7605	7648	7692	7736	7779	44
995	99- 7823	7867	791	7954	7998	8041	8085	8129	8172	8216	44
996	99- 8259	8303	8347	839	8434	8477	8521	8564	8608	8652	44
997	99- 8695	8739	8782	8826	8869	8913	8956	9	9043	9087	44
998	99- 9131	9174	9218	9261	9305	9348	9392	9435	9479	9522	44
999	99- 9565	9609	9652	9696	9739	9783	9826	987	9913	9957	43
No.	0	1	2	3	4	5	6	7	8	9	D

Hyperbolic Logarithms.

No.	Log.	No.	Log.	No.	Log.	No.	Log.
1	0.0000	3.8	1.3350	7.6	2.0281	17	2.8332
1.05	0.0488	3.9	1.3610	7.7	2.0412	17.5	2.8621
1.1	0.0953	4	1.3863	7.8	2.0541	18	2.8904
1.15	0.1398	4.1	1.4110	7.9	2.0669	18.5	2.9178
1.2	0.1823	4.2	1.4351	8	2.0794	19	2.9444
1.25	0.2231	4.3	1.4586	8.1	2.0919	19.5	2.9703
1.3	0.2624	4.4	1.4816	8.2	2.1041	20	2.9957
1.35	0.3001	4.5	1.5041	8.3	2.1163	21	3.0445
1.4	0.3365	4.6	1.5261	8.4	2.1282	22	3.0910
1.45	0.3716	4.7	1.5476	8.5	2.1401	23	3.1355
1.5	0.4055	4.8	1.5686	8.6	2.1518	24	3.1781
1.55	0.4383	4.9	1.5892	8.7	2.1633	25	3.2189
1.6	0.4700	5	1.6094	8.8	2.1748	26	3.2581
1.65	0.5008	5.1	1.6292	8.9	2.1861	27	3.2958
1.7	0.5306	5.2	1.6487	9	2.1972	28	3.3322
1.75	0.5596	5.3	1.6677	9.1	2.2083	29	3.3673
1.8	0.5878	5.4	1.6864	9.2	2.2192	30	3.4012
1.85	0.6152	5.5	1.7047	9.3	2.2300	31	3.4340
1.9	0.6419	5.6	1.7228	9.4	2.2407	32	3.4657
1.95	0.6678	5.7	1.7405	9.5	2.2513	33	3.4965
2	0.6931	5.8	1.7579	9.6	2.2618	34	3.5263
2.1	0.7419	5.9	1.7750	9.7	2.2721	35	3.5553
2.2	0.7885	6	1.7918	9.8	2.2824	36	3.5835
2.3	0.8329	6.1	1.8083	9.9	2.2925	37	3.6109
2.4	0.8755	6.2	1.8245	10	2.3026	38	3.6376
2.5	0.9163	6.3	1.8405	10.5	2.3513	39	3.6636
2.6	0.9555	6.4	1.8563	11	2.3979	40	3.6889
2.7	0.9933	6.5	1.8718	11.5	2.4430	41	3.7136
2.8	1.0296	6.6	1.8871	12	2.4849	42	3.7377
2.9	1.0647	6.7	1.9021	12.5	2.5262	43	3.7612
3	1.0986	6.8	1.9169	13	2.5649	44	3.7842
3.1	1.1312	6.9	1.9315	13.5	2.6027	45	3.8067
3.2	1.1632	7	1.9459	14	2.6391	46	3.8286
3.3	1.1939	7.1	1.9601	14.5	2.6740	47	3.8501
3.4	1.2238	7.2	1.9741	15	2.7081	48	3.8712
3.5	1.2528	7.3	1.9879	15.5	2.7408	49	3.8918
3.6	1.2809	7.4	2.0015	16	2.7726	50	3.9120
3.7	1.3083	7.5	2.0149	16.5	2.8034		

Squares and Cubes, Square Roots and Cube Roots for 1 to 1,000.

No.	Square.	Cube.	Sq. Rt.	C. Rt.	No.	Square.	Cube.	Sq. Rt.	C. Rt.
1	1	1	1.0000	1.0000	61	3721	226981	7.8102	3.9365
2	4	8	1.4142	1.2599	62	3844	238328	7.8740	3.9579
3	9	27	1.7321	1.4422	63	3969	250047	7.9373	3.9791
4	16	64	2.0000	1.5874	64	4096	262144	8.0000	4.
5	25	125	2.2361	1.7100	65	4225	274625	8.0623	4.0207
6	36	216	2.4495	1.8171	66	4356	287496	8.1240	4.0412
7	49	343	2.6458	1.9129	67	4489	300763	8.1854	4.0615
8	64	512	2.8284	2.0000	68	4624	314432	8.2462	4.0817
9	81	729	3.0000	2.0801	69	4761	328509	8.3066	4.1016
10	100	1000	3.1623	2.1544	70	4900	343000	8.3666	4.1213
11	121	1331	3.3166	2.2240	71	5041	357911	8.4261	4.1408
12	144	1728	3.4641	2.2894	72	5184	373248	8.4853	4.1602
13	169	2197	3.6056	2.3513	73	5329	389017	8.5440	4.1793
14	196	2744	3.7417	2.4101	74	5476	405224	8.6023	4.1983
15	225	3375	3.8730	2.4662	75	5625	421875	8.6603	4.2172
16	256	4096	4.0000	2.5198	76	5776	438976	8.7178	4.2358
17	289	4913	4.1231	2.5713	77	5929	456533	8.7750	4.2543
18	324	5832	4.2426	2.6207	78	6084	474552	8.8318	4.2727
19	361	6859	4.3589	2.6684	79	6241	493039	8.8882	4.2908
20	400	8000	4.4721	2.7144	80	6400	512000	8.9443	4.3089
21	441	9261	4.5826	2.7589	81	6561	531441	9.	4.3267
22	484	10648	4.6904	2.8020	82	6724	551368	9.0554	4.3445
23	529	12167	4.7958	2.8439	83	6889	571787	9.1104	4.3621
24	576	13824	4.8990	2.8845	84	7056	592704	9.1652	4.3795
25	625	15625	5.0000	2.9240	85	7225	614125	9.2195	4.3968
26	676	17576	5.0990	2.9625	86	7396	636056	9.2736	4.4140
27	729	19683	5.1962	3.0000	87	7569	658503	9.3274	4.4310
28	784	21952	5.2915	3.0366	88	7744	681472	9.3808	4.4480
29	841	24389	5.3852	3.0723	89	7921	704969	9.4340	4.4647
30	900	27000	5.4772	3.1072	90	8100	729000	9.4868	4.4814
31	961	29791	5.5678	3.1414	91	8281	753571	9.5394	4.4979
32	1024	32768	5.6569	3.1748	92	8464	778688	9.5917	4.5144
33	1089	35937	5.7446	3.2075	93	8649	804357	9.6437	4.5307
34	1156	39304	5.8310	3.2396	94	8836	830584	9.6954	4.5468
35	1225	42875	5.9161	3.2711	95	9025	857375	9.7468	4.5628
36	1296	46656	6.0000	3.3019	96	9216	884736	9.7980	4.5789
37	1369	50653	6.0828	3.3322	97	9409	912673	9.8489	4.5947
38	1444	54872	6.1644	3.3620	98	9604	941192	9.8995	4.6104
39	1521	59319	6.2450	3.3912	99	9801	970299	9.9499	4.6261
40	1600	64000	6.3246	3.4200	100	10000	1000000	10.	4.6416
41	1681	68921	6.4031	3.4482	101	10201	1030301	10.0499	4.6570
42	1764	74088	6.4807	3.4760	102	10404	1061208	10.0995	4.6723
43	1849	79507	6.5574	3.5034	103	10609	1092727	10.1489	4.6875
44	1936	85184	6.6332	3.5303	104	10816	1124864	10.1980	4.7027
45	2025	91125	6.7082	3.5569	105	11025	1157625	10.2470	4.7177
46	2116	97336	6.7823	3.5830	106	11236	1191016	10.2956	4.7326
47	2209	103823	6.8557	3.6088	107	11449	1225043	10.3441	4.7475
48	2304	110592	6.9282	3.6342	108	11664	1259712	10.3923	4.7622
49	2401	117649	7.0000	3.6593	109	11881	1295029	10.4403	4.7769
50	2500	125000	7.0711	3.6840	110	12100	1331000	10.4881	4.7914
51	2601	132651	7.1414	3.7084	111	12321	1367631	10.5357	4.8059
52	2704	140608	7.2111	3.7325	112	12544	1404928	10.5830	4.8203
53	2809	148877	7.2801	3.7563	113	12769	1442897	10.6301	4.8346
54	2916	157464	7.3485	3.7798	114	12996	1481544	10.6771	4.8488
55	3025	166375	7.4162	3.8030	115	13225	1520875	10.7238	4.8629
56	3136	175616	7.4833	3.8259	116	13456	1560896	10.7703	4.8770
57	3249	185193	7.5498	3.8485	117	13689	1601613	10.8167	4.8910
58	3364	195112	7.6158	3.8709	118	13924	1643032	10.8628	4.9049
59	3481	205379	7.6811	3.8930	119	14161	1685159	10.9087	4.9187
60	3600	216000	7.7460	3.9149	120	14400	1728000	10.9545	4.9324

Squares and Cubes, Square Roots and Cube Roots

(Continued.)

No.	Square.	Cube.	Sq. Rt.	C. Rt.	No.	Square.	Cube.	Sq. Rt.	C. Rt.
121	14641	1771561	11.	4.9461	186	34596	6434856	13.6382	5.7083
122	14884	1815848	11.0454	4.9597	187	34969	6539203	13.6748	5.7185
123	15129	1860867	11.0905	4.9732	188	35344	6644672	13.7113	5.7287
124	15376	1906624	11.1355	4.9866	189	35721	6751269	13.7477	5.7388
125	15625	1953125	11.1803	5.	190	36100	6859000	13.7840	5.7489
126	15876	2000376	11.2250	5.0133	191	36481	6967871	13.8203	5.7590
127	16129	2048383	11.2694	5.0265	192	36864	7077888	13.8564	5.7690
128	16384	2097152	11.3137	5.0397	193	37249	7189057	13.8924	5.7790
129	16641	2146689	11.3578	5.0528	194	37636	7301384	13.9284	5.7890
130	16900	2197000	11.4018	5.0658	195	38025	7414875	13.9642	5.7989
131	17161	2248091	11.4455	5.0788	196	38416	7529536	14.	5.8088
132	17424	2299968	11.4891	5.0916	197	38809	7645373	14.0357	5.8186
133	17689	2352637	11.5326	5.1045	198	39204	7762392	14.0712	5.8285
134	17956	2406104	11.5758	5.1172	199	39601	7880599	14.1067	5.8383
135	18225	2460375	11.6190	5.1299	200	40000	8000000	14.1421	5.8480
136	18496	2515456	11.6619	5.1426	201	40401	8120601	14.1774	5.8578
137	18769	2571353	11.7047	5.1551	202	40804	8242408	14.2127	5.8675
138	19044	2628072	11.7473	5.1676	203	41209	8365427	14.2478	5.8771
139	19321	2685619	11.7898	5.1801	204	41616	8489664	14.2829	5.8868
140	19600	2744000	11.8322	5.1925	205	42025	8615125	14.3178	5.8964
141	19881	2803221	11.8743	5.2048	206	42436	8741816	14.3527	5.9059
142	20164	2863288	11.9164	5.2171	207	42849	8869743	14.3875	5.9155
143	20449	2924207	11.9583	5.2293	208	43264	8998912	14.4222	5.9250
144	20736	2985984	12.	5.2415	209	43681	9129329	14.4568	5.9345
145	21025	3048625	12.0416	5.2536	210	44100	9261000	14.4914	5.9439
146	21316	3112136	12.0830	5.2656	211	44521	9393931	14.5258	5.9533
147	21609	3176523	12.1244	5.2776	212	44944	9528128	14.5602	5.9627
148	21904	3241792	12.1655	5.2896	213	45369	9663597	14.5945	5.9721
149	22201	3307949	12.2066	5.3015	214	45796	9800344	14.6287	5.9814
150	22500	3375000	12.2474	5.3133	215	46225	9938375	14.6629	5.9907
151	22801	3442951	12.2882	5.3251	216	46656	10077696	14.6969	6.
152	23104	3511808	12.3288	5.3368	217	47089	10218313	14.7309	6.0092
153	23409	3581577	12.3693	5.3485	218	47524	10360232	14.7648	6.0185
154	23716	3652264	12.4097	5.3601	219	47961	10503459	14.7986	6.0277
155	24025	3723875	12.4499	5.3717	220	48400	10648000	14.8324	6.0368
156	24336	3796416	12.4900	5.3832	221	48841	10793861	14.8661	6.0459
157	24649	3869893	12.5300	5.3947	222	49284	10941048	14.8997	6.0550
158	24964	3944312	12.5698	5.4061	223	49729	11089567	14.9332	6.0641
159	25281	4019679	12.6095	5.4175	224	50176	11239424	14.9666	6.0732
160	25600	4096000	12.6491	5.4288	225	50625	11390625	15.	6.0822
161	25921	4173281	12.6886	5.4401	226	51076	11543176	15.0333	6.0912
162	26244	4251528	12.7279	5.4514	227	51529	11697083	15.0665	6.1002
163	26569	4330747	12.7671	5.4626	228	51984	11852352	15.0997	6.1091
164	26896	4410944	12.8062	5.4737	229	52441	12008989	15.1327	6.1180
165	27225	4492125	12.8452	5.4848	230	52900	12167000	15.1658	6.1269
166	27556	4574296	12.8841	5.4959	231	53361	12326391	15.1987	6.1358
167	27889	4657463	12.9228	5.5069	232	53824	12487168	15.2315	6.1446
168	28224	4741632	12.9615	5.5178	233	54289	12649337	15.2643	6.1534
169	28561	4826809	13.	5.5288	234	54756	12812904	15.2971	6.1622
170	28900	4913000	13.0384	5.5397	235	55225	12977875	15.3297	6.1710
171	29241	5000211	13.0767	5.5505	236	55696	13144256	15.3623	6.1797
172	29584	5088448	13.1149	5.5613	237	56169	13312053	15.3948	6.1885
173	29929	5177717	13.1529	5.5721	238	56644	13481272	15.4272	6.1972
174	30276	5268024	13.1909	5.5828	239	57121	13651919	15.4596	6.2058
175	30625	5359375	13.2288	5.5934	240	57600	13824000	15.4919	6.2145
176	30976	5451776	13.2665	5.6041	241	58081	13997521	15.5242	6.2231
177	31329	5545233	13.3041	5.6147	242	58564	14172488	15.5563	6.2317
178	31684	5639752	13.3417	5.6252	243	59049	14348907	15.5885	6.2402
179	32041	5735339	13.3791	5.6357	244	59536	14526784	15.6205	6.2488
180	32400	5832000	13.4164	5.6462	245	60025	14706125	15.6525	6.2573
181	32761	5929741	13.4536	5.6567	246	60516	14886936	15.6844	6.2658
182	33124	6028568	13.4907	5.6671	247	61009	15069223	15.7162	6.2743
183	33489	6128487	13.5277	5.6774	248	61504	15252992	15.7480	6.2828
184	33856	6229504	13.5647	5.6877	249	62001	15438249	15.7797	6.2912
185	34225	6331625	13.6015	5.6980	250	62500	15625000	15.8114	6.2996

Squares and Cubes, Square Roots and Cube Roots

(Continued.)

No.	Square.	Cube.	Sq. Rt.	C. Rt	No.	Square.	Cube.	Sq. Rt.	C. Rt.
251	63001	15813251	15.8430	6.3030	316	99856	31554496	17.7764	6.8113
252	63504	16003008	15.8745	6.3164	317	100489	31855013	17.8045	6.8165
253	64009	16194277	15.9060	6.3217	318	101124	32157432	17.8326	6.8256
254	64516	16387064	15.9374	6.3330	319	101761	32461759	17.8606	6.8328
255	65025	16581375	15.9687	6.3413	320	102400	32768000	17.8885	6.8389
256	65536	16777216	16.	6.3496	321	103041	33076161	17.9165	6.8470
257	66049	16974593	16.0312	6.3579	322	103684	33386248	17.9444	6.8541
258	66564	17173512	16.0624	6.3661	323	104329	33698267	17.9722	6.8612
259	67081	17373979	16.0935	6.3743	324	104976	34012224	18.	6.8683
260	67600	17576000	16.1245	6.3825	325	105625	34328125	18.0278	6.8753
261	68121	17779581	16.1555	6.3907	326	106276	34645976	18.0555	6.8824
262	68644	17984728	16.1864	6.3988	327	106929	34965783	18.0831	6.8894
263	69169	18191447	16.2173	6.4070	328	107584	35287552	18.1108	6.8964
264	69696	18399744	16.2481	6.4151	329	108241	35611289	18.1384	6.9034
265	70225	18609625	16.2788	6.4232	330	108900	35937000	18.1659	6.9104
266	70756	18821096	16.3095	6.4312	331	109561	36264691	18.1934	6.9174
267	71289	19034163	16.3401	6.4393	332	110224	36594368	18.2209	6.9244
268	71824	19248832	16.3707	6.4473	333	110889	36926037	18.2483	6.9313
269	72361	19465109	16.4012	6.4553	334	111556	37259704	18.2757	6.9383
270	72900	19683000	16.4317	6.4633	335	112225	37595375	18.3030	6.9451
271	73441	19902511	16.4621	6.4713	336	112896	37933056	18.3303	6.9521
272	73984	20123648	16.4924	6.4792	337	113569	38272753	18.3576	6.9589
273	74529	20346417	16.5227	6.4872	338	114244	38614472	18.3848	6.9658
274	75076	20570824	16.5529	6.4951	339	114921	38958219	18.4120	6.9727
275	75625	20796875	16.5831	6.5030	340	115600	39304000	18.4391	6.9795
276	76176	21024576	16.6132	6.5108	341	116281	39651821	18.4662	6.9864
277	76729	21253933	16.6433	6.5187	342	116964	40001688	18.4932	6.9932
278	77284	21484952	16.6733	6.5265	343	117649	40353607	18.5203	7.
279	77841	21717639	16.7033	6.5343	344	118336	40707584	18.5472	7.0068
280	78400	21952000	16.7332	6.5421	345	119025	41063625	18.5742	7.0136
281	78961	22188041	16.7631	6.5499	346	119716	41421736	18.6011	7.0203
282	79524	22425768	16.7929	6.5577	347	120409	41781923	18.6279	7.0271
283	80089	22665187	16.8226	6.5654	348	121104	42144192	18.6548	7.0338
284	80656	22906304	16.8523	6.5731	349	121801	42508549	18.6815	7.0406
285	81225	23149125	16.8819	6.5808	350	122500	42875000	18.7083	7.0473
286	81796	23393656	16.9115	6.5885	351	123201	43243551	18.7350	7.0540
287	82369	23639903	16.9411	6.5962	352	123904	43614208	18.7617	7.0607
288	82944	23887872	16.9706	6.6039	353	124609	43986977	18.7883	7.0674
289	83521	24137569	17.	6.6115	354	125316	44361864	18.8149	7.0740
290	84100	24389000	17.0294	6.6191	355	126025	44738875	18.8414	7.0807
291	84681	24642171	17.0587	6.6267	356	126736	45118016	18.8680	7.0873
292	85264	24897088	17.0880	6.6343	357	127449	45499293	18.8944	7.0940
293	85849	25153757	17.1172	6.6419	358	128164	45882712	18.9209	7.1006
294	86436	25412184	17.1464	6.6494	359	128881	46268279	18.9473	7.1072
295	87025	25672375	17.1756	6.6569	360	129600	46656000	18.9737	7.1138
296	87616	25934336	17.2047	6.6644	361	130321	47045881	19.	7.1204
297	88209	26198073	17.2337	6.6719	362	131044	47437928	19.0263	7.1269
298	88804	26463592	17.2627	6.6794	363	131769	47832147	19.0526	7.1335
299	89401	26730899	17.2916	6.6869	364	132496	48228544	19.0788	7.1400
300	90000	27000000	17.3205	6.6943	365	133225	48627125	19.1050	7.1466
301	90601	27270901	17.3494	6.7018	366	133956	49027896	19.1311	7.1531
302	91204	27543608	17.3781	6.7092	367	134689	49430863	19.1572	7.1596
303	91809	27818127	17.4069	6.7166	368	135424	49836032	19.1833	7.1661
304	92416	28094464	17.4356	6.7240	369	136161	50243409	19.2094	7.1726
305	93025	28372625	17.4642	6.7313	370	136900	50653000	19.2354	7.1791
306	93636	28652616	17.4929	6.7387	371	137641	51064811	19.2614	7.1855
307	94249	28934443	17.5214	6.7460	372	138384	51478848	19.2873	7.1920
308	94864	29218112	17.5499	6.7533	373	139129	51895117	19.3132	7.1984
309	95481	29503629	17.5784	6.7606	374	139876	52313624	19.3391	7.2048
310	96100	29791000	17.6068	6.7679	375	140625	52734375	19.3649	7.2112
311	96721	30080231	17.6352	6.7752	376	141376	53157376	19.3907	7.2177
312	97344	30371328	17.6635	6.7824	377	142129	53582633	19.4165	7.2240
313	97969	30664297	17.6918	6.7897	378	142884	54010152	19.4422	7.2304
314	98596	30959144	17.7200	6.7969	379	143641	54439939	19.4679	7.2368
315	99225	31255875	17.7482	6.8041	380	144400	54872000	19.4936	7.2432

Squares and Cubes, Square Roots and Cube Roots

(Continued.)

No.	Square.	Cube.	Sq. Rt.	C. Rt.	No.	Square.	Cube.	Sq. Rt.	C. Rt.
391	145161	56800311	19.5192	7.3195	446	198916	88716536	21.1187	7.6403
392	145924	57153968	19.5448	7.3258	447	199809	89314623	21.1424	7.6460
393	146689	57512457	19.5704	7.3322	448	200704	89915392	21.1660	7.6517
394	147456	57876104	19.5959	7.3385	449	201601	90518849	21.1896	7.6574
395	148225	57066625	19.6214	7.3748	450	202500	91125000	21.2132	7.6631
396	148996	57512456	19.6469	7.3811	451	203401	91733851	21.2368	7.6688
397	149769	57960603	19.6723	7.3874	452	204304	92345408	21.2603	7.6744
398	150544	58411072	19.6977	7.3936	453	205209	92959677	21.2838	7.6801
399	151321	58863869	19.7231	7.3999	454	206116	93576664	21.3073	7.6857
400	152100	59319000	19.7484	7.4061	455	207025	94196375	21.3307	7.6914
401	152881	59776471	19.7737	7.4124	456	207936	94818816	21.3542	7.6970
402	153664	60236288	19.7990	7.4186	457	208849	95443993	21.3776	7.7026
403	154449	60698457	19.8242	7.4248	458	209764	96071912	21.4009	7.7082
404	155236	61162984	19.8494	7.4310	459	210681	96702579	21.4243	7.7138
405	156025	61629875	19.8746	7.4372	460	211600	97336000	21.4476	7.7194
406	156816	62099136	19.8997	7.4434	461	212521	97972181	21.4709	7.7250
407	157609	62570773	19.9249	7.4496	462	213444	98611128	21.4942	7.7306
408	158404	63044792	19.9499	7.4558	463	214369	99252847	21.5174	7.7362
409	159201	63521199	19.9750	7.4619	464	215296	99897344	21.5407	7.7418
410	160000	64000000	20.	7.4681	465	216225	100544625	21.5639	7.7473
411	160801	64481201	20.0250	7.4742	466	217156	101194696	21.5870	7.7529
412	161604	64964808	20.0499	7.4803	467	218089	101847563	21.6102	7.7584
413	162409	65450827	20.0749	7.4864	468	219024	102503232	21.6333	7.7639
414	163216	65939264	20.0998	7.4925	469	219961	103161709	21.6564	7.7695
415	164025	66430125	20.1246	7.4986	470	220900	103823000	21.6795	7.7750
416	164836	66923416	20.1494	7.5047	471	221841	104487111	21.7025	7.7805
417	165649	67419143	20.1742	7.5108	472	222784	105154048	21.7256	7.7860
418	166464	67917312	20.1990	7.5169	473	223729	105823817	21.7486	7.7915
419	167281	68417929	20.2237	7.5229	474	224676	106496424	21.7715	7.7970
410	168100	68921000	20.2485	7.5290	475	225625	107171875	21.7945	7.8025
411	168921	69426531	20.2731	7.5350	476	226576	107850176	21.8174	7.8079
412	169744	69934528	20.2978	7.5410	477	227529	108531333	21.8403	7.8134
413	170569	70444997	20.3224	7.5470	478	228484	109215352	21.8632	7.8188
414	171396	70957944	20.3470	7.5530	479	229441	109902239	21.8861	7.8243
415	172225	71473375	20.3715	7.5590	480	230400	110592000	21.9089	7.8297
416	173056	71991296	20.3961	7.4650	481	231361	111284641	21.9317	7.8352
417	173889	72511713	20.4206	7.4710	482	232324	111980168	21.9545	7.8406
418	174724	73034632	20.4450	7.4770	483	233289	112678587	21.9773	7.8460
419	175561	73560059	20.4695	7.4829	484	234256	113379904	22.	7.8514
420	176400	74088000	20.4939	7.4889	485	235225	114084125	22.0227	7.8568
421	177241	74618461	20.5183	7.4948	486	236196	114791256	22.0454	7.8622
422	178084	75151448	20.5426	7.5007	487	237169	115501303	22.0681	7.8676
423	178929	75686967	20.5670	7.5067	488	238144	116214272	22.0907	7.8730
424	179776	76225024	20.5913	7.5126	489	239121	116930169	22.1133	7.8784
425	180625	76765625	20.6155	7.5185	490	240100	117649000	22.1359	7.8837
426	181476	77308776	20.6398	7.5244	491	241081	118370771	22.1585	7.8891
427	182329	77854483	20.6640	7.5302	492	242064	119095488	22.1811	7.8944
428	183184	78402752	20.6882	7.5361	493	243049	119823157	22.2036	7.8998
429	184041	78953589	20.7123	7.5420	494	244036	120553784	22.2261	7.9051
430	184900	79507000	20.7364	7.5478	495	245025	121287375	22.2486	7.9105
431	185761	80062991	20.7605	7.5537	496	246016	122023936	22.2711	7.9158
432	186624	80621568	20.7846	7.5595	497	247009	122763473	22.2935	7.9211
433	187489	81182737	20.8087	7.5654	498	248004	123505992	22.3159	7.9264
434	188356	81746504	20.8327	7.5712	499	249001	124251499	22.3383	7.9317
435	189225	82312875	20.8567	7.5770	500	250000	125000000	22.3607	7.9370
436	190096	82881856	20.8806	7.5828	501	251001	125751501	22.3830	7.9423
437	190969	83453453	20.9045	7.5886	502	252004	126506008	22.4054	7.9476
438	191844	84027672	20.9284	7.5944	503	253009	127263527	22.4277	7.9528
439	192721	84604519	20.9523	7.6001	504	254016	128024064	22.4499	7.9581
440	193600	85184000	20.9761	7.6059	505	255025	128787625	22.4722	7.9634
441	194481	85766121	21.	7.6117	506	256036	129554216	22.4944	7.9686
442	195364	86350888	21.0238	7.6174	507	257049	130323843	22.5167	7.9739
443	196249	86938307	21.0476	7.6232	508	258064	131096512	22.5390	7.9791
444	197136	87528384	21.0713	7.6289	509	259081	131872229	22.5610	7.9843
445	198025	88121125	21.0950	7.6346	510	260100	132651000	22.5832	7.9896

Squares and Cubes, Square Roots and Cube Roots

(Continued.)

No.	Square.	Cube.	Sq. Rt.	C. Rt.	No.	Square.	Cube.	Sq. Rt.	C. Rt.
511	261121	133432831	22.6053	7.9948	576	331776	191102976	24.	8.3203
512	262144	134217728	22.6274	8.	577	332929	192100033	24.0208	8.3251
513	263169	135005697	22.6495	8.0052	578	334084	193100552	24.0416	8.3300
514	264196	135796744	22.6716	8.0104	579	335241	194104539	24.0624	8.3348
515	265225	136590875	22.6936	8.0156	580	336400	195112000	24.0832	8.3396
516	266256	137388096	22.7156	8.0208	581	337561	196122941	24.1039	8.3443
517	267289	138188413	22.7376	8.0260	582	338724	197137368	24.1247	8.3491
518	268324	138991832	22.7596	8.0311	583	339889	198155287	24.1454	8.3539
519	269361	139798359	22.7816	8.0363	584	341056	199176704	24.1661	8.3587
520	270400	140608000	22.8035	8.0415	585	342225	200201625	24.1868	8.3634
521	271441	141420761	22.8254	8.0466	586	343396	201230056	24.2074	8.3682
522	272484	142236648	22.8473	8.0517	587	344569	202262003	24.2281	8.3730
523	273529	143055667	22.8692	8.0569	588	345744	203297472	24.2487	8.3777
524	274576	143877824	22.8910	8.0620	589	346921	204336469	24.2693	8.3825
525	275625	144703125	22.9129	8.0671	590	348100	205379000	24.2899	8.3872
526	276676	145531576	22.9347	8.0723	591	349281	206425071	24.3105	8.3919
527	277729	146363183	22.9565	8.0774	592	350464	207474688	24.3311	8.3967
528	278784	147197952	22.9783	8.0825	593	351649	208527857	24.3516	8.4014
529	279841	148035889	23.	8.0876	594	352836	209584584	24.3721	8.4061
530	280900	148877000	23.0217	8.0927	595	354025	210644875	24.3926	8.4108
531	281961	149721291	23.0434	8.0978	596	355216	211708736	24.4131	8.4155
532	283024	150568768	23.0651	8.1028	597	356409	212776173	24.4336	8.4202
533	284089	151419437	23.0868	8.1079	598	357604	213847192	24.4540	8.4249
534	285156	152273304	23.1084	8.1130	599	358801	214921799	24.4745	8.4296
535	286225	153130375	23.1301	8.1180	600	360000	216000000	24.4949	8.4343
536	287296	153990656	23.1517	8.1231	601	361201	217081801	24.5153	8.4390
537	288369	154854153	23.1733	8.1281	602	362404	218167208	24.5357	8.4437
538	289444	155720872	23.1948	8.1332	603	363609	219256227	24.5561	8.4484
539	290521	156590819	23.2164	8.1382	604	364816	220348864	24.5764	8.4530
540	291600	157464000	23.2379	8.1433	605	366025	221445125	24.5967	8.4577
541	292681	158340421	23.2594	8.1483	606	367236	222545016	24.6171	8.4623
542	293764	159220088	23.2809	8.1533	607	368449	223648543	24.6374	8.4670
543	294849	160103007	23.3024	8.1583	608	369664	224755712	24.6577	8.4716
544	295936	160989184	23.3238	8.1633	609	370881	225866529	24.6779	8.4763
545	297025	161878625	23.3452	8.1683	610	372100	226981000	24.6982	8.4809
546	298116	162771336	23.3666	8.1733	611	373321	228099131	24.7184	8.4856
547	299209	163667323	23.3880	8.1783	612	374544	229220928	24.7386	8.4902
548	300304	164566592	23.4094	8.1833	613	375769	230346397	24.7588	8.4948
549	301401	165469149	23.4307	8.1882	614	376996	231475544	24.7790	8.4994
550	302500	166375000	23.4521	8.1932	615	378225	232608375	24.7992	8.5040
551	303601	167284151	23.4734	8.1982	616	379456	233744896	24.8193	8.5086
552	304704	168196608	23.4947	8.2031	617	380689	234885113	24.8395	8.5132
553	305809	169112377	23.5160	8.2081	618	381924	236029032	24.8596	8.5178
554	306916	170031464	23.5372	8.2130	619	383161	237176659	24.8797	8.5224
555	308025	170953875	23.5584	8.2180	620	384400	238328000	24.8998	8.5270
556	309136	171879616	23.5797	8.2229	621	385641	239483061	24.9199	8.5316
557	310249	172808693	23.6008	8.2278	622	386884	240641848	24.9399	8.5362
558	311364	173741112	23.6220	8.2327	623	388129	241804367	24.9600	8.5408
559	312481	174676879	23.6432	8.2377	624	389376	242970624	24.9800	8.5453
560	313600	175616000	23.6643	8.2426	625	390625	244140625	25.	8.5499
561	314721	176558481	23.6854	8.2475	626	391876	245314376	25.0200	8.5544
562	315844	177504328	23.7065	8.2524	627	393129	246491883	25.0400	8.5590
563	316969	178453547	23.7276	8.2573	628	394384	247673152	25.0599	8.5635
564	318096	179406144	23.7487	8.2621	629	395641	248858189	25.0799	8.5681
565	319225	180362125	23.7697	8.2670	630	396900	250047000	25.0998	8.5726
566	320356	181321496	23.7908	8.2719	631	398161	251239591	25.1197	8.5772
567	321489	182284263	23.8118	8.2768	632	399424	252435968	25.1396	8.5817
568	322624	183250432	23.8328	8.2816	633	400689	253636137	25.1595	8.5862
569	323761	184220009	23.8537	8.2865	634	401956	254840104	25.1794	8.5907
570	324900	185193000	23.8747	8.2913	635	403225	256047875	25.1992	8.5952
571	326041	186169411	23.8956	8.2962	636	404496	257259456	25.2190	8.5997
572	327184	187149248	23.9165	8.3010	637	405769	258474853	25.2389	8.6043
573	328329	188132517	23.9374	8.3059	638	407044	259694072	25.2587	8.6088
574	329476	189119224	23.9583	8.3107	639	408321	260917119	25.2784	8.6132
575	330625	190109375	23.9792	8.3155	640	409600	262144000	25.2982	8.6177

Squares and Cubes, Square Roots and Cube Roots

(Continued.)

No.	Square.	Cube.	Sq. Rt.	C. Rt.	No.	Square.	Cube.	Sq. Rt.	C. Rt.
641	410881	263374721	25.3180	8.6222	706	498436	351895816	26.5707	8.9043
642	412164	264609288	25.3377	8.6287	707	499849	353393243	26.5895	8.9085
643	413449	265847707	25.3574	8.6311	708	501264	354894912	26.6083	8.9127
644	414736	267089984	25.3772	8.6357	709	502681	356400829	26.6271	8.9169
645	416025	268336125	25.3969	8.6401	710	504100	357911000	26.6458	8.9211
646	417316	269586136	25.4165	8.6446	711	505521	359425431	26.6646	8.9253
647	418609	270840023	25.4362	8.6490	712	506944	360944128	26.6833	8.9295
648	419904	272097792	25.4558	8.6535	713	508369	362467097	26.7021	8.9337
649	421201	273359449	25.4755	8.6579	714	509796	363994344	26.7208	8.9378
650	422500	274625000	25.4951	8.6624	715	511225	365525875	26.7395	8.9420
651	423801	275894451	25.5147	8.6668	716	512656	367061696	26.7582	8.9462
652	425104	277167808	25.5343	8.6713	717	514089	368601813	26.7769	8.9503
653	426409	278445077	25.5539	8.6757	718	515524	370146232	26.7955	8.9545
654	427716	279726264	25.5734	8.6801	719	516961	371694959	26.8142	8.9587
655	429025	281011375	25.5930	8.6845	720	518400	373248000	26.8328	8.9628
656	430336	282300416	25.6125	8.6890	721	519841	374805361	26.8514	8.9670
657	431649	283593393	25.6320	8.6934	722	521284	376367048	26.8701	8.9711
656	432964	284890312	25.6515	8.6978	723	522729	377933067	26.8887	8.9752
659	434281	286191179	25.6710	8.7022	724	524176	379503424	26.9072	8.9794
660	435600	287496000	25.6905	8.7066	725	525625	381078125	26.9258	8.9835
661	436921	288804781	25.7099	8.7110	726	527076	382657176	26.9444	8.9876
662	438244	290117528	25.7294	8.7154	727	528529	384240583	26.9629	8.9918
663	439569	291434247	25.7488	8.7198	728	529984	385828352	26.9815	8.9959
664	440896	292754944	25.7682	8.7241	729	531441	387420489	27.	9.
665	442225	294079625	25.7876	8.7285	730	532900	389017000	27.0185	9.0041
666	443556	295408296	25.8070	8.7329	731	534361	390617891	27.0370	9.0082
667	444889	296740963	25.8263	8.7373	732	535824	392223168	27.0555	9.0123
668	446224	298077632	25.8457	8.7416	733	537289	393832837	27.0740	9.0164
669	447561	299418309	25.8650	8.7460	734	538756	395446904	27.0924	9.0205
670	448900	300763000	25.8844	8.7503	735	540225	397065375	27.1109	9.0246
671	450241	302111711	25.9037	8.7547	736	541696	398688256	27.1293	9.0287
672	451584	303464448	25.9230	8.7590	737	543169	400315553	27.1477	9.0328
673	452929	304821217	25.9422	8.7634	738	544644	401947272	27.1662	9.0369
674	454276	306182024	25.9615	8.7677	739	546121	403583419	27.1846	9.0410
675	455625	307546875	25.9808	8.7721	740	547600	405224000	27.2029	9.0450
676	456976	308915776	26.	8.7764	741	549081	406869021	27.2213	9.0491
677	458329	310288733	26.0192	8.7807	742	550564	408518488	27.2397	9.0532
678	459684	311665752	26.0384	8.7850	743	552049	410172407	27.2580	9.0572
679	461041	313046839	26.0576	8.7893	744	553536	411830784	27.2764	9.0613
680	462400	314432000	26.0768	8.7937	745	555025	413493625	27.2947	9.0654
681	463761	315821241	26.0960	8.7980	746	556516	415160936	27.3130	9.0694
682	465124	317214568	26.1151	8.8023	747	558009	416832723	27.3313	9.0735
683	466489	318611987	26.1343	8.8066	748	559504	418508992	27.3496	9.0775
684	467856	320013504	26.1534	8.8109	749	561001	420189749	27.3679	9.0816
685	469225	321419125	26.1725	8.8152	750	562500	421875000	27.3861	9.0856
686	470596	322828856	26.1916	8.8194	751	564001	423564751	27.4044	9.0896
687	471969	324242703	26.2107	8.8237	752	565504	425259008	27.4226	9.0937
688	473344	325660672	26.2298	8.8280	753	567009	426957777	27.4408	9.0977
689	474721	327082769	26.2488	8.8323	754	568516	428661064	27.4591	9.1017
690	476100	328509000	26.2679	8.8366	755	570025	430368875	27.4773	9.1057
691	477481	329939371	26.2869	8.8408	756	571536	432081216	27.4955	9.1098
692	478864	331373888	26.3059	8.8451	757	573049	433798093	27.5136	9.1138
693	480249	332812557	26.3249	8.8493	758	574564	435519512	27.5318	9.1178
694	481636	334255384	26.3439	8.8536	759	576081	437245479	27.5500	9.1218
695	483025	335702375	26.3629	8.8578	760	577600	438976000	27.5681	9.1258
696	484416	337153536	26.3818	8.8621	761	579121	440711081	27.5862	9.1298
697	485809	338608873	26.4008	8.8663	762	580644	442450728	27.6043	9.1338
698	487204	340068392	26.4197	8.8706	763	582169	444194947	27.6225	9.1378
699	488601	341532099	26.4386	8.8748	764	583696	445943744	27.6405	9.1418
700	490000	343000000	26.4575	8.8790	765	585225	447697125	27.6586	9.1458
701	491401	344472101	26.4764	8.8833	766	586756	449455096	27.6767	9.1498
702	492804	345948408	26.4953	8.8875	767	588289	451217663	27.6948	9.1537
703	494209	347428927	26.5141	8.8917	768	589824	452984832	27.7128	9.1577
704	495616	348913664	26.5330	8.8959	769	591361	454756609	27.7308	9.1617
705	497025	350402625	26.5518	8.9001	770	592900	456533000	27.7489	9.1657

Squares and Cubes, Square Roots and Cube Roots

(Continued.)

No.	Square.	Cube.	Sq. Rt.	C. Rt.	No.	Square.	Cube.	Sq. Rt.	C. Rt.
771	594441	458314011	27.7669	9.1696	836	698896	584277056	28.9157	9.4394
772	595984	460099648	27.7849	9.1736	837	700569	586078253	28.9310	9.4341
773	597529	461889917	27.8029	9.1775	838	702244	587880172	28.9482	9.4279
774	599076	463684824	27.8209	9.1815	839	703921	589687719	28.9655	9.4316
775	600625	465484375	27.8388	9.1855	840	705600	591704000	28.9828	9.4364
776	602176	467288576	27.8568	9.1894	841	707281	594323321	29.	9.4391
777	603729	469097433	27.8747	9.1933	842	708964	596947688	29.0172	9.4439
778	605284	470910952	27.8927	9.1973	843	710649	599077107	29.0345	9.4486
779	606841	472729139	27.9106	9.2012	844	712336	601211584	29.0517	9.4533
780	608400	474552000	27.9285	9.2052	845	714025	603351125	29.0689	9.4541
781	609961	476379541	27.9464	9.2091	846	715716	605495736	29.0861	9.4578
782	611524	478211768	27.9643	9.2130	847	717409	607645423	29.1033	9.4615
783	613089	480048687	27.9821	9.2170	848	719104	609800192	29.1204	9.4652
784	614656	481890304	28.	9.2209	849	720801	611960049	29.1376	9.4689
785	616225	483736625	28.0179	9.2248	850	722500	614125000	29.1548	9.4727
786	617796	485587656	28.0357	9.2287	851	724201	616295051	29.1719	9.4764
787	619369	487443403	28.0535	9.2326	852	725904	618470208	29.1890	9.4801
788	620944	489303872	28.0713	9.2365	853	727609	620650477	29.2062	9.4838
789	622521	491169069	28.0891	9.2404	854	729316	622835864	29.2233	9.4875
790	624100	493039000	28.1069	9.2443	855	731025	625026375	29.2404	9.4912
791	625681	494913671	28.1247	9.2482	856	732736	627222016	29.2575	9.4949
792	627264	496793088	28.1425	9.2521	857	734449	629422793	29.2746	9.4986
793	628849	498677257	28.1603	9.2560	858	736164	631628712	29.2916	9.5023
794	630436	500566184	28.1780	9.2599	859	737881	633839779	29.3087	9.5060
795	632025	502459875	28.1957	9.2638	860	739600	636056000	29.3258	9.5097
796	633616	504358336	28.2135	9.2677	861	741321	638277381	29.3428	9.5134
797	635209	506261573	28.2312	9.2716	862	743044	640503928	29.3598	9.5171
798	636804	508169592	28.2489	9.2754	863	744769	642735647	29.3769	9.5207
799	638401	510082399	28.2666	9.2793	864	746496	644972544	29.3939	9.5244
800	640000	512000000	28.2843	9.2832	865	748225	647214625	29.4109	9.5281
801	641601	513922401	28.3019	9.2870	866	749956	649461896	29.4279	9.5317
802	643204	515849608	28.3196	9.2909	867	751689	651714363	29.4449	9.5354
803	644809	517781627	28.3373	9.2948	868	753424	653972032	29.4618	9.5391
804	646416	519718464	28.3549	9.2986	869	755161	656234909	29.4788	9.5427
805	648025	521660125	28.3725	9.3025	870	756900	658503000	29.4958	9.5464
806	649636	523606616	28.3901	9.3063	871	758641	660776311	29.5127	9.5501
807	651249	525557943	28.4077	9.3102	872	760384	663054848	29.5296	9.5537
808	652864	527514112	28.4253	9.3140	873	762129	665338617	29.5466	9.5574
809	654481	529475129	28.4429	9.3179	874	763876	667627624	29.5636	9.5610
810	656100	531441000	28.4605	9.3217	875	765625	669921875	29.5804	9.5647
811	657721	533411731	28.4781	9.3255	876	767376	672221376	29.5973	9.5683
812	659344	535387328	28.4956	9.3294	877	769129	674526133	29.6142	9.5719
813	660969	537367797	28.5132	9.3333	878	770884	676836152	29.6311	9.5756
814	662596	539353144	28.5307	9.3370	879	772641	679151439	29.6479	9.5792
815	664225	541343375	28.5482	9.3408	880	774400	681472000	29.6648	9.5828
816	665856	543338496	28.5657	9.3447	881	776161	683797841	29.6816	9.5865
817	667489	545338513	28.5832	9.3485	882	777924	686128968	29.6985	9.5901
818	669124	547343432	28.6007	9.3523	883	779689	688465387	29.7153	9.5937
819	670761	549353259	28.6182	9.3561	884	781456	690807104	29.7321	9.5972
820	672400	551368000	28.6356	9.3599	885	783225	693154125	29.7489	9.6010
821	674041	553387661	28.6531	9.3637	886	784996	695506456	29.7658	9.6046
822	675684	555412248	28.6705	9.3676	887	786769	697864103	29.7825	9.6082
823	677329	557441767	28.6880	9.3713	888	788544	700227072	29.7993	9.6118
824	678976	559476224	28.7054	9.3751	889	790321	702596369	29.8161	9.6154
825	680625	561515625	28.7228	9.3789	890	792100	704969000	29.8329	9.6190
826	682276	563559976	28.7402	9.3827	891	793881	707347971	29.8496	9.6226
827	683929	565609283	28.7576	9.3865	892	795664	709732288	29.8664	9.6262
828	685584	567663552	28.7750	9.3902	893	797449	712121957	29.8831	9.6298
829	687241	569722789	28.7924	9.3940	894	799236	714516984	29.8998	9.6334
830	688900	571787000	28.8097	9.3978	895	801025	716917375	29.9166	9.6370
831	690561	573856191	28.8271	9.4016	896	802816	719323136	29.9333	9.6406
832	692224	575930368	28.8444	9.4053	897	804609	721734273	29.9500	9.6442
833	693889	578009537	28.8617	9.4091	898	806404	724150792	29.9666	9.6477
834	695556	580093704	28.8791	9.4129	899	808201	726572699	29.9833	9.6513
835	697225	582182875	28.8964	9.4166	900	810000	729000000	30.	9.6549

Squares and Cubes, Square Roots and Cube Roots

(Continued.)

No.	Square.	Cube.	Sq. Rt.	C. Rt.	No.	Square.	Cube.	Sq. Rt.	C. Rt.
901	811801	731432701	30.0167	9.6585	951	904401	860085351	30.8383	9.8339
902	813604	733870808	30.0333	9.6620	952	906304	863101408	30.8545	9.8374
903	815409	736314327	30.0500	9.6656	953	908209	866529177	30.8707	9.8408
904	817216	738763264	30.0666	9.6692	954	910116	869258664	30.8869	9.8443
905	819025	741217625	30.0832	9.6727	955	912025	870983875	30.9031	9.8477
906	820836	743677416	30.0998	9.6763	956	913936	873722816	30.9192	9.8511
907	822649	746142643	30.1164	9.6799	957	915849	876467493	30.9354	9.8546
908	824464	748613312	30.1330	9.6834	958	917764	879217912	30.9516	9.8580
909	826281	751089429	30.1496	9.6870	959	919681	881974079	30.9677	9.8614
910	828100	753571000	30.1662	9.6905	960	921600	884736000	30.9839	9.8648
911	829921	756058031	30.1828	9.6941	961	923521	887503681	31.	9.8683
912	831744	758550528	30.1993	9.6976	962	925444	890277128	31.0161	9.8717
913	833569	761048497	30.2159	9.7012	963	927369	893056347	31.0322	9.8751
914	835396	763551944	30.2324	9.7047	964	929296	895841344	31.0483	9.8785
915	837225	766060875	30.2490	9.7082	965	931225	898632125	31.0644	9.8819
916	839056	768575296	30.2655	9.7118	966	933156	901428696	31.0805	9.8854
917	840889	771095213	30.2820	9.7153	967	935089	904231063	31.0966	9.8888
918	842724	773620632	30.2985	9.7188	968	937024	907039232	31.1127	9.8922
919	844561	776151559	30.3150	9.7224	969	938961	909853209	31.1288	9.8956
920	846400	778688000	30.3315	9.7259	970	940900	912673000	31.1448	9.8990
921	848241	781229961	30.3480	9.7294	971	942841	915498611	31.1609	9.9024
922	850084	783777448	30.3645	9.7329	972	944784	918330048	31.1769	9.9058
923	851929	786330467	30.3809	9.7364	973	946729	921167317	31.1929	9.9092
924	853776	788889024	30.3974	9.7400	974	948676	924010424	31.2090	9.9126
925	855625	791453125	30.4138	9.7435	975	950625	926859375	31.2250	9.9160
926	857476	794022776	30.4302	9.7470	976	952576	929714176	31.2410	9.9194
927	859329	796597983	30.4467	9.7505	977	954529	932574833	31.2570	9.9227
928	861184	799178752	30.4631	9.7540	978	956484	935441352	31.2730	9.9261
929	863041	801765089	30.4795	9.7575	979	958441	938313739	31.2890	9.9295
930	864900	804357000	30.4959	9.7610	980	960400	941192000	31.3050	9.9329
931	866761	806954491	30.5123	9.7645	981	962361	944076141	31.3209	9.9363
932	868624	809557568	30.5287	9.7680	982	964324	946966168	31.3369	9.9396
933	870489	812166237	30.5450	9.7715	983	966289	949862087	31.3528	9.9430
934	872356	814780504	30.5614	9.7750	984	968256	952763904	31.3688	9.9464
935	874225	817400375	30.5778	9.7785	985	970225	955671625	31.3847	9.9497
936	876096	820025856	30.5941	9.7819	986	972196	958585256	31.4006	9.9531
937	877969	822656953	30.6105	9.7854	987	974169	961504803	31.4166	9.9565
938	879844	825293672	30.6268	9.7889	988	976144	964430272	31.4325	9.9598
939	881721	827936019	30.6431	9.7924	989	978121	967361669	31.4484	9.9632
940	883600	830584000	30.6594	9.7959	990	980100	970299000	31.4643	9.9666
941	885481	833237621	30.6757	9.7993	991	982081	973242271	31.4802	9.9699
942	887364	835896888	30.6920	9.8028	992	984064	976191488	31.4960	9.9733
943	889249	838561807	30.7083	9.8063	993	986049	979146657	31.5119	9.9766
944	891136	841232384	30.7246	9.8097	994	988036	982107784	31.5278	9.9800
945	893025	843908625	30.7409	9.8132	995	990025	985074875	31.5436	9.9833
946	894916	846590536	30.7571	9.8167	996	992016	988047936	31.5595	9.9866
947	896809	849278123	30.7734	9.8201	997	994009	991026973	31.5753	9.9900
948	898704	851971392	30.7896	9.8236	998	996004	994011992	31.5911	9.9933
949	900601	854670349	30.8058	9.8270	999	998001	997002999	31.6070	9.9967
950	902500	857375000	30.8221	9.8303	1000	1000000	1000000000	31.6228	10.

To Extract a Higher Root Than the Cube.

The fourth root is the square root of the square root. The sixth root is the cube root of the square root or the square root of the cube root.

For square roots and cube roots see tables under their respective heads.

Other roots are most conveniently found by the use of logarithms.

Decimal Equivalents for Each Sixty-Fourth.

32nds.	64ths.	Decimal.	Fraction.	32nds.	64ths.	Decimal.	Fraction.
	1	.015625			33	.515625	
1	2	.03125		17	34	.53125	
	3	.046875			35	.546875	
2	4	.0625	1/16	18	36	.5625	9/16
	5	.078125			37	.578125	
3	6	.09375		19	38	.59375	
	7	.109365			39	.609375	
4	8	.125	1/8	20	40	.625	5/8
	9	.140625			41	.640625	
5	10	.15625		21	42	.65625	
	11	.171875			43	.671875	
6	12	.1875	3/16	22	44	.6875	11/16
	13	.203125			45	.703125	
7	14	.21875		23	46	.71875	
	15	.234375			47	.734375	
8	16	.25	1/4	24	48	.75	3/4
	17	.265625			49	.765625	
9	18	.28125		25	50	.78125	
	19	.296875			51	.796875	
10	20	.3125	5/16	26	52	.8125	13/16
	21	.328125			53	.828125	
11	22	.34375		27	54	.84375	
	23	.359375			55	.859375	
12	24	.375	3/8	28	56	.875	7/8
	25	.390625			57	.890625	
13	26	.40625		29	58	.90625	
	27	.421875			59	.921875	
14	28	.4375	7/16	30	60	.9375	15/16
	29	.453125			61	.953125	
15	30	.46875		31	62	.96875	
	31	.484375			63	.984375	
16	32	.5	1/2	32	64	1.	1

Diameters, Circumferences and Areas of Circles (1 to 150 Diameter).

Diam.	Circum.	Area.	Diam.	Circum.	Area.	Diam.	Circum.	Area.
1	3.1416	0.7854	51	160.22	2042.82	101	317.30	8011.85
2	6.2832	3.1416	52	163.36	2123.72	102	320.44	8171.28
3	9.4248	7.0686	53	166.50	2206.18	103	323.58	8332.29
4	12.5664	12.5664	54	169.65	2290.22	104	326.73	8494.87
5	15.7080	19.635	55	172.79	2375.83	105	329.87	8659.01
6	18.850	28.274	56	175.93	2463.01	106	333.01	8824.73
7	21.991	38.485	57	179.07	2551.76	107	336.15	8992.02
8	25.133	50.266	58	182.21	2642.08	108	339.29	9160.88
9	28.274	63.617	59	185.35	2733.97	109	342.43	9331.32
10	31.416	78.540	60	188.50	2827.43	110	345.58	9503.32
11	34.558	95.033	61	191.64	2922.47	111	348.72	9676.89
12	37.699	113.10	62	194.78	3019.07	112	351.86	9852.03
13	40.841	132.73	63	197.92	3117.25	113	355.00	10028.75
14	43.982	153.94	64	201.06	3216.99	114	358.14	10207.03
15	47.124	176.71	65	204.20	3318.31	115	361.28	10386.89
16	50.265	201.06	66	207.34	3421.19	116	364.42	10568.32
17	53.407	226.98	67	210.49	3525.65	117	367.57	10751.32
18	56.549	254.47	68	213.63	3631.68	118	370.71	10935.88
19	59.690	283.53	69	216.77	3739.28	119	373.85	11122.02
20	62.832	314.16	70	219.91	3848.45	120	376.99	11309.73
21	65.973	346.36	71	223.05	3959.19	121	380.13	11499.01
22	69.115	380.13	72	226.19	4071.50	122	383.27	11689.87
23	72.257	415.48	73	229.34	4185.39	123	386.42	11882.29
24	75.398	452.39	74	232.48	4300.84	124	389.56	12076.28
25	78.540	490.87	75	235.62	4417.86	125	392.70	12271.85
26	81.681	530.93	76	238.76	4536.46	126	395.84	12468.98
27	84.823	572.56	77	241.90	4656.63	127	398.98	12667.69
28	87.965	615.75	78	245.04	4778.36	128	402.12	12867.96
29	91.106	660.52	79	248.19	4901.67	129	405.27	13069.81
30	94.248	706.86	80	251.33	5026.55	130	408.41	13273.23
31	97.389	754.77	81	254.47	5153.00	131	411.55	13478.22
32	100.53	804.25	82	257.61	5281.02	132	414.69	13684.78
33	103.67	855.30	83	260.75	5410.61	133	417.83	13892.91
34	106.81	907.92	84	263.89	5541.77	134	420.97	14102.61
35	109.96	962.11	85	267.04	5674.50	135	424.12	14313.88
36	113.10	1017.88	86	270.18	5808.80	136	427.26	14526.72
37	116.24	1075.21	87	273.32	5944.68	137	430.40	14741.14
38	119.38	1134.11	88	276.46	6082.12	138	433.54	14957.12
39	122.52	1194.59	89	279.60	6221.14	139	436.68	15174.68
40	125.66	1256.64	90	282.74	6361.73	140	439.82	15393.80
41	128.81	1320.25	91	285.88	6503.88	141	442.96	15614.50
42	131.95	1385.44	92	289.03	6647.61	142	446.11	15836.77
43	135.09	1452.20	93	292.17	6792.91	143	449.25	16060.61
44	138.23	1520.53	94	295.31	6939.78	144	452.39	16286.02
45	141.37	1590.43	95	298.45	7088.22	145	455.53	16513.00
46	144.51	1661.90	96	301.59	7238.23	146	458.67	16741.55
47	147.65	1734.94	97	304.73	7389.81	147	461.81	16971.67
48	150.80	1809.56	98	307.88	7542.96	148	464.96	17203.36
49	153.94	1885.74	99	311.02	7697.69	149	468.10	17436.62
50	157.08	1963.50	100	314.16	7853.98	150	471.24	17671.46

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(In the alphabetical arrangement of subjects, smaller words such as "of," "in," etc., are neglected. The letter T., wherever it appears, is an abbreviation for "table.")

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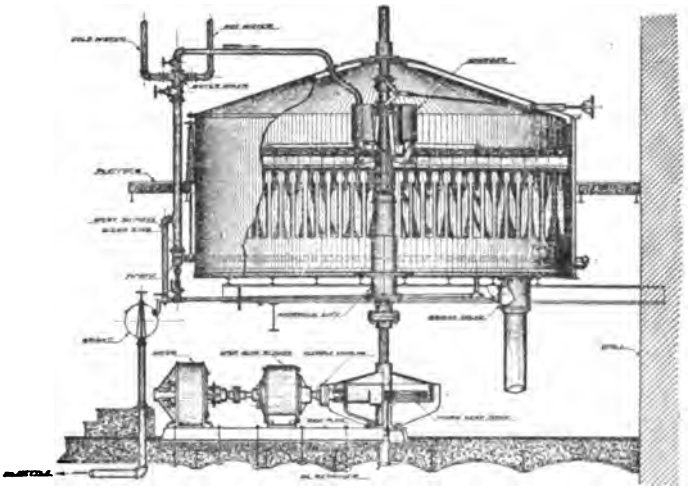

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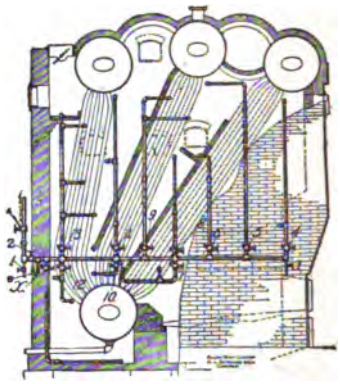

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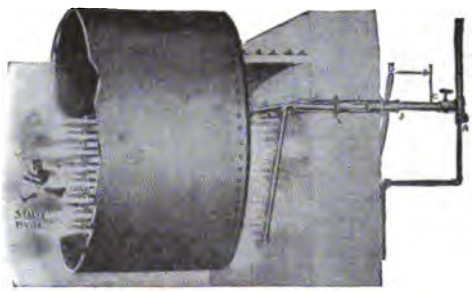

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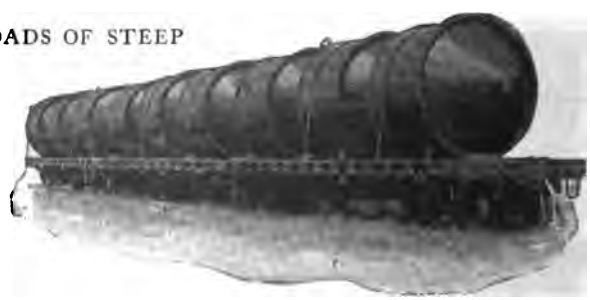

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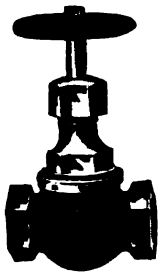

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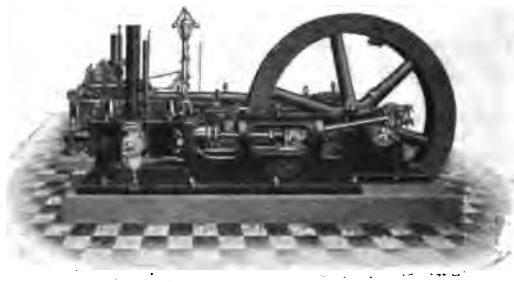

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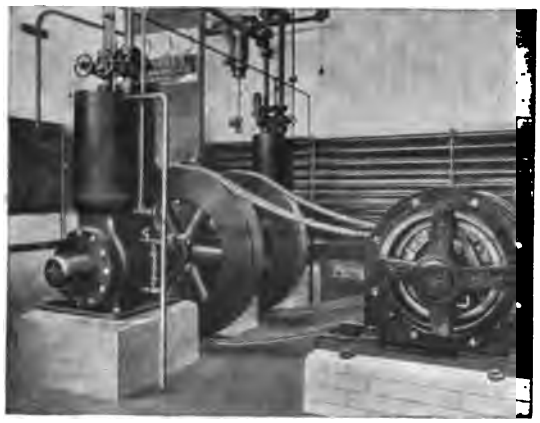

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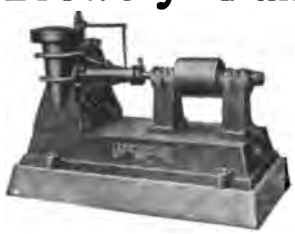

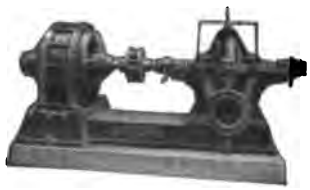

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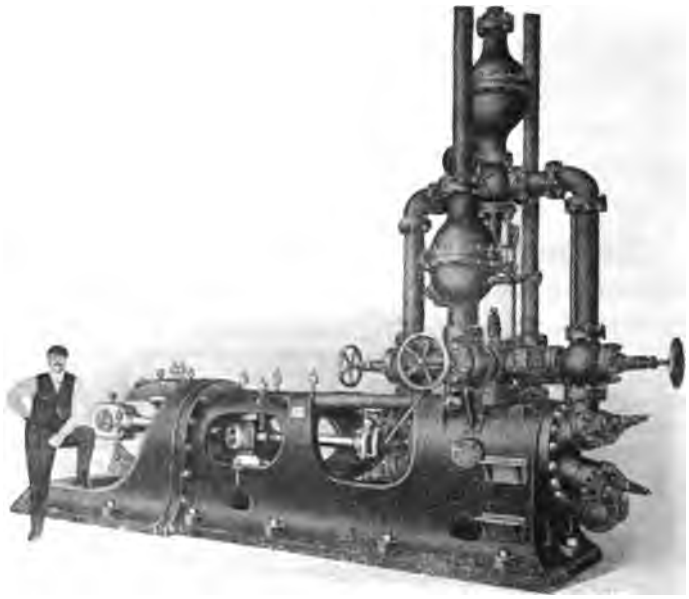

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